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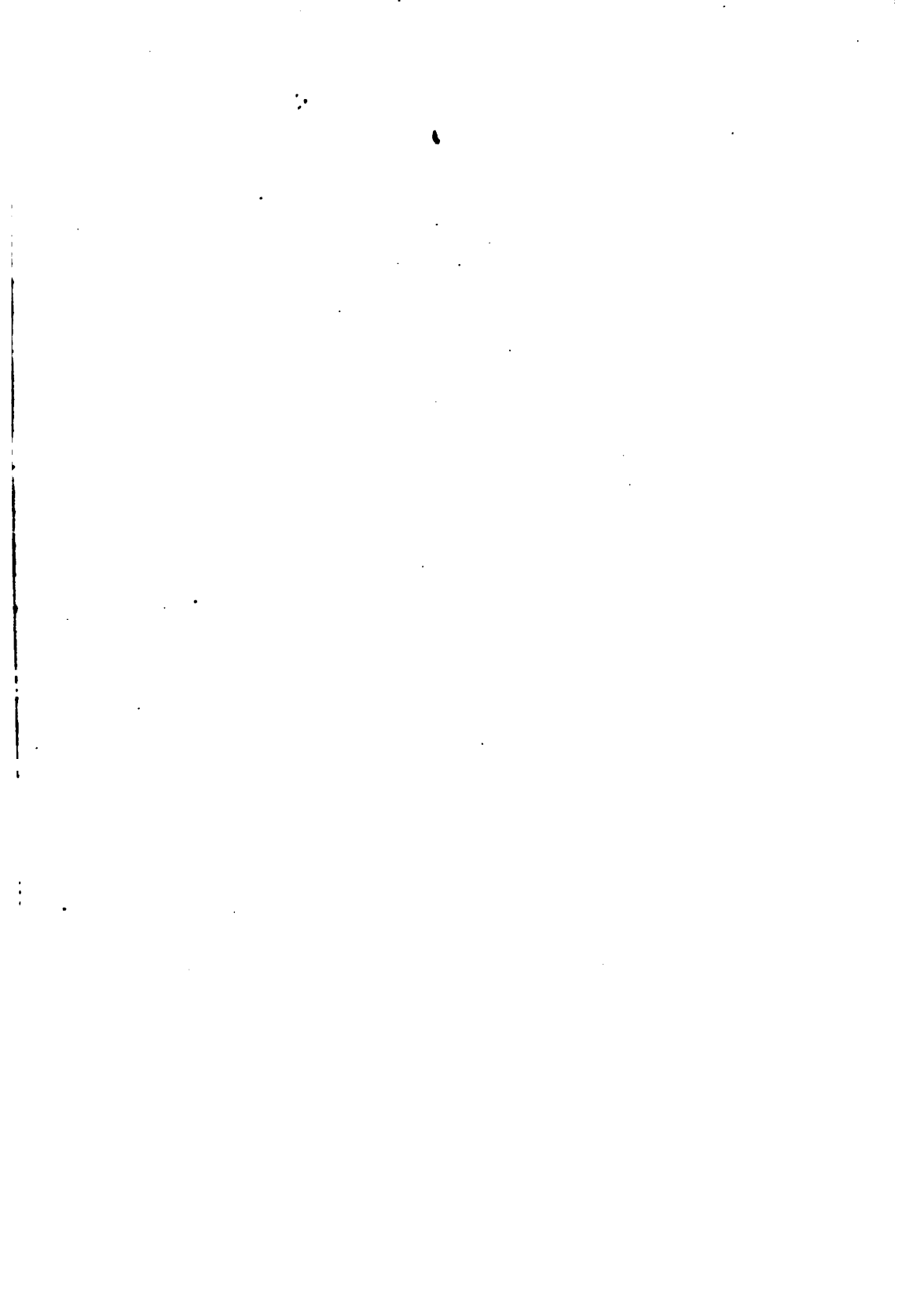
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DESIGN

OF A

HIGH SPEED STEAM ENGINE.

Notes, Diagrams, Formulas and Tables.

~~PART I.~~

Part 2 of this work has since been issued as
a separate & independent work.

BY
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NOTES, DIAGRAMS, FORMULAS AND TABLES

FOR THE DESIGN OF A

HIGH SPEED STEAM ENGINE.

DATA.

Determine the principal dimensions of an engine fulfilling the following requirements:

The work ordinarily required, = 100 *H.P.* net = 110 *I.H.P.*, but the load may vary from 50 *I.H.P.* to 170 *I.H.P.*

The engine is to run steadily under all these loads, the total variation ($N_1 - N_2$) of the number of revolutions of engine shaft being less than $\frac{1}{50}$ of the normal number of revolutions (N)

$$i.e., N_1 - N_2 = \frac{N}{50} \text{ when } N = \frac{N_1 + N_2}{2}. \quad (1)$$

The engine is to be of the non-condensing, high (rotative) speed type, durable, of compact and rigid form, and the steam consumption of the 100 *H.P.* per hour is to be a minimum. Moreover, this high speed engine must be provided with generous wearing surfaces which are continuously lubricated, the reciprocating parts must be balanced and their weights so chosen that the tangential pressures on the crank pin will be as nearly constant as possible.

It is assumed that the workmanship will be excellent, the parallelism of crank pin and shaft, as well as the alignment of crank-shaft bearings, perfect, and that they will be maintained so by good foundations.

ORDER OF CALCULATION.

- I. Determination of diagram of effective steam pressures on piston of required engine.
 - a. Determination of boiler pressure.
 - b. Determination of clearance, real cut-off and apparent cut-off.
 - c. Determination of back pressure and amount of compression.
 - d. Method of constructing equilateral hyperbola and of determining points on the curves $pv^n = \text{constant}$ by tabulated coordinates.
 - e. Methods of determining the mean effective pressures of indicator diagrams.
 - f. Determination of the cut-off corresponding to a given horse power H_1 when the mean effective pressure p_m and the total pressure p''_m is known for another horse power H of the same engine running at the same speed, with same initial pressure p_1 and same counterpressures.
 - g. Diagrams of effective-steam-pressure-on-piston for forward and return strokes.
- II. Determination of ratio of length of connecting rod to that of crank.
- III. Determination of that mean accelerating force (per $\square''$ of piston) necessary to start reciprocating parts and corresponding to the most uniform tangential pressure on crank pin.
 - a. Determination of acceleration and construction of diagrams of accelerating forces.

- b.* Construction of horizontal or resultant pressures on cross-head pin (or on crank pin when friction is neglected) by combining diagram of effective steam pressures on piston with diagram of acceleration.
 - c.* Discussion of effects on this diagram of valve setting, high speed and long and short cut-offs.
 - d.* Conversion of horizontal pressures on crank pin into tangential pressures.
 - e.* Construction of diagrams of tangential pressures on crank pin.
- IV. Determination of diameter of cylinder, length of stroke, revolutions per minute and weight of reciprocating parts.
 - V. Preliminary estimate of dimensions of reciprocating parts, to see whether minimum weight of latter does not exceed that necessary to produce steadiness of running under the conditions assumed.
 - VI. Determination of weight of fly wheel rim from tangential pressure diagram.
 - VII. Determination of dimensions of crank disk and distribution of its material so that both crank arm and reciprocating parts will be balanced.
 - VIII. Method of determining the influence, on the foregoing results, of the frictional resistances, of the weight of the rod and of the exact values of the forces of inertia.
 - IX. Calculation of width and diameter of belt pulley.
 - X. Graphical determination of diameter of crank shaft.
 - XI. Calculation of the length of crank shaft journal and graphical determination of the plane of division of its brasses.
 - XII. Calculation of dimensions of steam ports and pipes, also thickness of cylinder walls.
 - XIII. Valve diagrams and dimensions of valve and gear.
 - XIV. Calculation of governor.
 - XV. Drawing of details of engine.
 - XVI. Drawing of plan and elevation of complete engine.

PLATES TO BE DRAWN.

- I. Six Indicator Diagrams.
- II. Six Diagrams of effective steam pressures on piston.
- III. Diagrams of accelerating and retarding pressures by approximate and exact methods.
- IV. Six Diagrams of horizontal pressures on cross-head pin.
- V. Diagram for converting horizontal pressures on cross-head pin into tangential pressures on crank pin.
- VI. Six Diagrams of tangential pressures on crank pin.
- VII. Four Diagrams of the forces shaking engine bed, for as many different degrees of counter weighting.
- VIII. Distribution of material on crank disk, and determination of its center of gravity.
- IX. Determination of diameter of engine shaft by the methods of Graphical Statics.
- IX. Graphical determination of the direction of the resultant pressure on crank shaft bearing.
- X. Valve diagrams.
Plates containing details, plan and elevations of complete engine.

I A.

DETERMINATION OF BOILER PRESSURE.

If we consider the cylinders as perfect non-conductors of heat, and consider economy of steam only, we must, in accordance with the deductions of thermodynamics, make our initial steam pressures as high as possible, for by so doing we increase the range of temperature of the steam in the engine, and thereby increase the efficiency of the engine. This is clearly shown by the expression for the theoretical efficiency η of the engine

$$\eta = \frac{T_1 - T_2}{T_1} \quad (2)$$

T_1 being the absolute temperature of the steam at the initial absolute pressure p_1 , and T_2 the lowest absolute temperature

practicably possible for the engine, being in non-condensing engines $T_2 = 212^\circ + 459.4 = 671.4$ Fahr. T_1 increases very slowly with the higher pressures of steam, see Roentgen's Thermodynamics, pp. 672 and 674.

Even for ordinary (cast iron) conducting cylinders it is true (up to not very well defined limits) that economy in the use of steam increases slowly with the initial pressure. Thus according to Prof. Thurston the weight of steam per hour per *H.P.* should in good engines be not more than that given by the empirical formula

$$\frac{200}{\sqrt{p_1}} \text{ pounds} \quad (3)$$

and for the best practice with large engines, dry steam, high piston speed and good design, construction and management the consumption of steam should be not more than

$$\frac{150}{\sqrt{p_1}} \quad (4)$$

Sometimes the highest pressure which can be employed in an engine is prescribed by the strength of the boilers which are to furnish it with steam; at other times (with engines running under conditions in which the power occasionally sinks considerably below the average) a lower pressure (accompanied by a greater cut-off) than economy of fuel and steam would prescribe, must be employed in order that the engine may run at high speed with great steadiness and without reversing the direction of the horizontal pressures on crank pin. See limits to high speed, III, c.

The employment of very high pressures is also limited by the greater first cost of the boilers owing to the greater strength required, and the greater care which must be exercised in their construction. The engine also becomes somewhat more costly for the same reason. A practical limit is thus set to the general use of extremely high pressures.

In the present case we will start the calculation by assuming the initial boiler pressure employed by the best makers of high speed engines when designing and rating their engines, namely, 85 pounds gauge pressure.

I B.

DETERMINATION OF CLEARANCE, REAL CUT-OFF AND APPARENT CUT-OFF.

Clearance is the volume or space included between the piston (when the latter is at the end of its stroke) and the cylinder head, and also includes the steam passages. It is usually expressed as a fraction of the volume swept through by the piston, *i.e.*, if i represents the clearance, A the area of piston and S the stroke we have

$$i = \frac{\text{clearance volume}}{AS} \quad (5)$$

The clearance should be as small as possible, for when it is at all large it involves considerable loss by presenting a large space to be filled by fresh steam, which steam is not engaged in pushing back the piston when entering the cylinder, but in giving whirling motions to its own particles, though during expansion a portion of the energy of the steam which has filled the clearance spaces is utilized. On account of the loss occasioned by it, the clearance is sometimes called the hurtful space. Its hurtful influence can however be entirely removed by allowing the exhaust steam to be compressed till it reaches a pressure equal to the initial pressure. When this is done no fresh steam is needed to fill the clearance spaces at the beginning of the stroke; but as it is not always practicable (as will appear later) to compress the exhaust steam to so great an extent, it behooves the designer to diminish the clearance as much as possible.

The total amount of engine clearance employed in practice varies greatly with the type and size of engine employed. For example, in Corliss Engines the four separate valves employed

are placed close to the cylinder and the distance between end of piston and cylinder head left for variation of length of connecting rod (when the wear is taken up) is frequently not more than $\frac{1}{4}$ ", so that in these engines the clearance often does not exceed $\frac{1}{4}$ of one per cent. In high speed engines, however, the steam passages are usually very large, as it is very important to admit the steam promptly, and at full pressure. In these engines the clearance sometimes amounts to 14 per cent., but this is very large. We will for the present assume that the clearance in our high speed engine is as great as 7 per cent.

CUT-OFF.

See Fig. 3, p 16.

The volume of steam in the cylinder (on the live steam side of the piston) when the admission valve closes, divided by the volume of this steam (after it has expanded) when the exhaust valve opens, is called the *real* or *virtual cut-off* $= \frac{1}{r} = \epsilon$. It is always a proper fraction. The reciprocal of this fraction is the *expansion*, and is of course a whole or mixed number.

The distance of the piston from the end of its stroke when the admission valve closes, divided by the whole stroke, is the *apparent cut off* $= \frac{1}{r^1} = \epsilon^1$. The relation between the real and apparent cut-off is represented by the following equation :

$$\text{real cut-off} = \frac{1}{r} = \frac{\frac{SA}{r^1} + iSA}{SA + iSA} = \frac{\frac{1}{r^1} + i}{1 + i} = \epsilon. \quad (6)$$

The following tables, pp. 8 and 9, give this relation, and were taken from Du Bois' translation of Weisbach, Vol. II. If our cylinders were non-conductors, and economy of steam were the only element of economy to be considered, we know from thermodynamics that it would be wise to employ both high pressures and great expansions. Unfortunately our cylinders are conductors of heat, and as a consequence there is a very marked

TABLE I.
REAL CUT-OFF CORRESPONDING TO APPARENT CUT-OFF FOR DIFFERENT
FRACTIONS OF CLEARANCE.

Apparent cut-off.	Clearance.										Apparent cut-off.
	.01	.02	.03	.04	.05	.06	.07	.08	.09	.10	
.05	.059	.069	.078	.087	.095	.104	.112	.120	.128	.136	.05
.06	.069	.078	.087	.096	.105	.113	.122	.130	.138	.145	.06
.07	.079	.088	.097	.106	.114	.123	.131	.139	.147	.155	.07
.08	.089	.098	.107	.115	.124	.132	.140	.148	.156	.164	.08
.09	.099	.108	.117	.125	.133	.142	.150	.157	.165	.173	.09
.10	.109	.118	.126	.135	.143	.151	.159	.167	.174	.182	.10
.11	.119	.127	.136	.144	.152	.160	.168	.176	.183	.191	.11
.12	.129	.137	.146	.154	.162	.170	.178	.185	.193	.200	.12
.13	.139	.147	.155	.163	.171	.179	.187	.194	.202	.209	.13
.14	.149	.157	.165	.173	.181	.189	.196	.204	.211	.218	.14
.15	.158	.167	.175	.183	.190	.198	.206	.214	.220	.227	.15
.16	.168	.176	.184	.192	.200	.208	.215	.222	.229	.236	.16
.17	.178	.186	.194	.202	.210	.217	.224	.231	.239	.245	.17
.18	.188	.196	.204	.212	.219	.226	.234	.241	.248	.255	.18
.19	.198	.206	.214	.221	.229	.236	.243	.250	.257	.264	.19
.20	.208	.216	.223	.231	.238	.245	.252	.259	.266	.273	.20
.21	.218	.225	.233	.240	.248	.255	.262	.269	.275	.282	.21
.22	.228	.235	.243	.250	.257	.264	.271	.278	.284	.291	.22
.23	.238	.245	.252	.260	.267	.274	.280	.287	.294	.300	.23
.24	.248	.255	.262	.269	.276	.283	.290	.296	.303	.309	.24
.25	.257	.265	.272	.279	.286	.292	.299	.306	.312	.318	.25
.26	.267	.275	.282	.288	.295	.302	.308	.315	.321	.327	.26
.27	.277	.284	.291	.298	.305	.311	.318	.324	.330	.336	.27
.28	.287	.294	.301	.308	.314	.321	.327	.333	.339	.345	.28
.29	.297	.304	.311	.317	.324	.330	.336	.343	.349	.355	.29
.30	.307	.314	.320	.327	.333	.340	.346	.352	.358	.364	.30
.31	.317	.324	.330	.337	.343	.349	.355	.361	.367	.373	.31
.32	.327	.334	.340	.346	.352	.358	.364	.370	.376	.382	.32
.33	.337	.343	.350	.356	.362	.368	.374	.380	.385	.391	.33
.34	.347	.353	.359	.365	.371	.377	.383	.389	.395	.400	.34
.35	.356	.363	.369	.375	.381	.387	.393	.398	.404	.409	.35
.36	.366	.373	.379	.385	.390	.396	.402	.407	.413	.418	.36
.37	.376	.382	.388	.394	.400	.406	.411	.417	.422	.427	.37
.38	.386	.392	.398	.404	.410	.415	.421	.426	.431	.436	.38
.39	.396	.402	.408	.413	.419	.425	.430	.435	.440	.445	.39
.40	.406	.412	.417	.423	.429	.434	.439	.444	.450	.455	.40

TABLE I, (CONTINUED.)

REAL CUT-OFF CORRESPONDING TO APPARENT CUT-OFF FOR DIFFERENT FRACTIONS OF CLEARANCE.

Apparent Cut-off.	Clearance.										Apparent Cut-off.
	.01	.02	.03	.04	.05	.06	.07	.08	.09	.10	
.41	.416	.422	.427	.433	.438	.443	.449	.454	.459	.463	.41
.42	.426	.431	.437	.442	.448	.453	.458	.463	.468	.473	.42
.43	.436	.441	.447	.452	.457	.462	.467	.472	.477	.482	.43
.44	.446	.451	.456	.462	.467	.472	.477	.481	.486	.491	.44
.45	.455	.461	.466	.471	.476	.481	.486	.491	.495	.500	.45
.46	.465	.471	.475	.481	.486	.491	.495	.500	.505	.509	.46
.47	.475	.480	.485	.490	.495	.500	.505	.509	.514	.518	.47
.48	.485	.490	.495	.500	.505	.509	.514	.519	.523	.527	.48
.49	.495	.500	.505	.510	.514	.519	.523	.528	.532	.536	.49
.50	.505	.510	.515	.519	.524	.528	.533	.537	.541	.545	.50
.51	.515	.520	.524	.529	.533	.538	.542	.546	.550	.554	.51
.52	.525	.529	.534	.538	.543	.544	.551	.556	.560	.564	.52
.53	.535	.539	.544	.548	.552	.557	.561	.565	.569	.573	.53
.54	.545	.549	.553	.558	.562	.566	.570	.574	.578	.582	.54
.55	.554	.559	.563	.567	.571	.575	.579	.583	.587	.591	.55
.56	.564	.569	.573	.577	.581	.585	.589	.593	.596	.600	.56
.57	.574	.578	.583	.587	.590	.594	.598	.602	.606	.609	.57
.58	.584	.588	.592	.596	.600	.604	.607	.611	.615	.618	.58
.59	.594	.598	.602	.606	.610	.613	.617	.620	.624	.627	.59
.60	.604	.608	.612	.615	.619	.623	.626	.630	.633	.636	.60
.61	.614	.618	.621	.625	.629	.632	.636	.639	.642	.645	.61
.62	.624	.627	.631	.635	.638	.642	.645	.648	.651	.655	.62
.63	.634	.637	.641	.644	.648	.651	.654	.657	.661	.664	.63
.64	.644	.647	.650	.654	.657	.660	.664	.667	.670	.673	.64
.65	.653	.657	.660	.663	.667	.669	.673	.676	.679	.682	.65
.66	.663	.667	.670	.673	.676	.679	.682	.685	.688	.691	.66
.67	.673	.676	.680	.683	.686	.689	.692	.694	.697	.700	.67
.68	.683	.685	.689	.692	.695	.698	.701	.704	.706	.709	.68
.69	.693	.695	.699	.702	.705	.708	.710	.713	.716	.718	.69
.70	.703	.706	.709	.712	.714	.717	.720	.722	.725	.727	.70
.71	.713	.716	.718	.721	.724	.726	.728	.731	.734	.736	.71
.72	.723	.725	.728	.731	.733	.736	.738	.741	.743	.745	.72
.73	.733	.735	.738	.740	.743	.745	.748	.750	.752	.755	.73
.74	.743	.745	.748	.750	.752	.755	.756	.759	.761	.764	.74
.75	.752	.755	.757	.760	.762	.764	.766	.769	.771	.773	.75

loss arising from the condensation of the live steam as it enters the cylinder and comes in contact with the surfaces of the latter which have just been cooled by contact with, and giving up their heat to, the exhaust steam. The complexity of this interchange of heat is such that thus far it has not been satisfactorily followed by calculation, and we must depend upon experiment for our knowledge of what constitutes the most economical point of cut-off with respect to the consumption of steam or fuel. See appendix A for a theoretical formula giving the most economical point of cut-off. This cut-off may be determined by means of an empirical formula representing the result of extensive and carefully conducted experiments by Mr. Charles Emery, namely,

$$\epsilon = \frac{1}{r} = \frac{1}{1 + \frac{p_1}{22}} = \frac{22}{22 + p_1} \quad (7)$$

ϵ = real cut-off. r = ratio of expansion. p_1 = absolute initial pressure.

Mr. Emery considers that the cut-offs given by this formula are "nearly correct for single engines of ordinary construction, and too large for the better class of compound engine." The first two columns of Table IV on page 18 contain the initial absolute pressure p_1 for various values of ϵ = real cut-off when $p_1 v^{1.1} = \text{constant}$ represents the curve of expansion. Thus far the only element of economy considered in choosing the point of cut-off was economy of steam or of the fuel which produced it. But this is only one of the elements which influences the cost per hour of the *H. P.* employed. Other elements such as wages of attendants, interest on cost of engine and boilers, repairs on engine and boilers, insurance, depreciation, etc., influence the current cost of the *H. P.* employed. That cut-off which makes this cost a minimum is evidently the true economical point of cut-off, the one in which the engine owner is financially interested. But as this cost, per horse-power per day, does not vary much for a considerable range of cut-off, we will here consider only the principal factor in that cost, *i. e.*, steam.

We may make a sufficient approximation to the correct cut-off by taking a value of the latter greater than that given by Mr. Emery's empirical formula for greatest economy of steam or fuel. for it is evident from general principles that the influence of the other expenses will be to make the proper cut-off greater than that corresponding to the greatest economy of steam or fuel. As our initial pressure was above assumed to be 85 pounds (by gauge) we get from Emery's formula $\frac{1}{r} < \frac{1}{5}$. If we now assume $\frac{1}{r} = \frac{1}{4}$ we shall have, both as regards the pressure and the cut-off, conditions such as are assumed by the best high speed engine builders.

I c.

DETERMINATION OF BACK PRESSURE AND AMOUNT OF COMPRESSION.

For well-designed non-condensing engines the back pressure during the exhaust is usually taken at 16 pounds (above a vacuum) per $\square''$, and in condensing engines at 2 pounds per $\square''$. As regards the amount of compression or cushioning suitable for high speed engines it may be said that while on the one hand compressing up to boiler pressure has the advantage of obviating the necessity of partly filling the clearance spaces with fresh steam at the beginning of each stroke, and also tends to diminish the condensation of this steam by heating up the cylinder walls, on the other hand it has the disadvantage of producing a reversal of pressure before the end of each stroke, which tends to produce a shock on the crank pin. If we wish to relieve the crank pin of all pressure at the end of the stroke we have only to compress the exhaust steam till its final pressure is equal to the sum of the terminal pressure (near end of the expansion line) and the pressure retarding the reciprocating parts. As in high speed engines the terminal pressure is practically reduced to a very small amount through expansion and an early exhaust, we may

vacuum lines, whatever the degree of expansion employed, draw the line OA and from the points A and C , where it cuts the steam and cut-off lines let fall the perpendiculars AB and CB . Their intersection at B will give a point of the curve exactly. For from the similar triangles AOD and OCE we have $CE : OE :: AD : OD$ and consequently $BD : OE :: FE : OD$. But OE and FE represent respectively the volume and pressure at the point of cut-off F , and OD and BD those at the point B , hence $p : v_1 :: p_1 : v$ or $pv = p_1v_1$, therefore, the point B was correctly found. Other points may be found in the same way.

METHOD OF DETERMINING POINTS, ON THE CURVES $pv^* =$ CONSTANT, BY TABULATED COORDINATES.

To find the coordinates for the curves by means of the following table we divide, in Fig. 2, the distance $HF = AC = v$, into

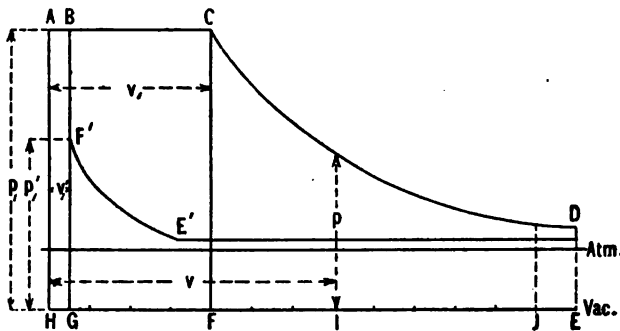


Fig. 2.

four equal parts and lay off one of these parts as many times as possible on FE . Then will for example $\frac{v}{v_1} = \frac{HI}{HF} = 1.75$, also $\frac{v^2}{v_1} = \frac{HJ}{FH} = 3$, etc. At the points of division on FE erect ordinates and lay off on them the value of $\frac{p}{p_1} \times p_1$ corresponding to the

TABLE II.

VALUES OF THE RATIO $\frac{p}{p_1}$ OF THE ABSOLUTE PRESSURES

OCCURRING IN FORMULA $p_1 v_1^n = p v^n = \text{CONSTANT}$.

FOR p IN LINEAR MEASURE MULTIPLY $\frac{p}{p_1}$ BY p_1 EXPRESSED IN LINEAR MEASURE.

		Exponent n in formula $p_1 v_1^n = p v^n$.						
$\frac{v}{v_1}$	Ratio of assumed vol. v to initial vol. v_1 .	Equilateral hyperbola curve for air expanding with temperature constant. $n = 1$.	$n = 1.0646 = \frac{1}{17}$ nearly. Curve for steam, dry satu- rated during expansion.	$n = 1.125 = \frac{9}{8}$. Adiabatic curve for steam with 10% moisture.	$n = 1.135$. Adiabatic curve for steam, dry saturated at begin- ning of expansion.	$n = 1.250 = \frac{5}{4}$. Compression curve for high speed engine with steam jacketed cylinder.	$n = 1.333 = \frac{4}{3}$. Adiabatic curve for super- heated steam.	$n = 1.408$. Adiabatic curve for air.
1.25	.8000	.7886	.7780	.7763	.7569	.7427	.7304	
1.50	.6667	.6494	.6337	.6312	.6024	.5824	.5650	
1.75	.5714	.5511	.5328	.5299	.4968	.4948	.4548	
2.00	.5000	.4781	.4585	.4550	.4265	.3969	.3767	
2.25	.4444	.4218	.4016	.3984	.3629	.3392	.3192	
2.50	.4000	.3779	.3567	.3535	.3121	.2947	.2752	
2.75	.3636	.3406	.3204	.3172	.2824	.2595	.2407	
3.00	.3333	.3105	.2906	.2874	.2533	.2311	.2129	
3.50	.2857	.2697	.2443	.2413	.2089	.1882	.1714	
4.00	.2500	.2286	.2102	.2073	.1768	.1575	.1420	
4.50	.2222	.2016	.1841	.1814	.1526	.1346	.1203	
5.	.2000	.1803	.1636	.1548	.1337	.1170	.1037	
6.	.1657	.1484	.1332	.1309	.1065	.0917	.0802	
7.	.1429	.1260	.1129	.1099	.0878	.0747	.0646	
8.	.1250	.1093	.0964	.0944	.0743	.0626	.0535	
9.	.1111	.0964	.0844	.0826	.0642	.0534	.0453	
10.	.1000	.0862	.0750	.0733	.0562	.0464	.0391	
11.	.0909	.0779	.0674	.0658	.0499	.0409	.0342	
12.	.0833	.0710	.0611	.0596	.0450	.0364	.0302	
13.	.0769	.0653	.0558	.0544	.0405	.0327	.0270	
14.	.0714	.0602	.0514	.0500	.0369	.0296	.0243	
15.	.0767	.0562	.0475	.0463	.0339	.0270	.0221	
16.	.0625	.0523	.0442	.0430	.0313	.0248	.0206	
18.	.0556	.0461	.0387	.0376	.0270	.0212	.0175	
20.	.0500	.0412	.0344	.0334	.0236	.0184	.0147	
22.	.0455	.0372	.0309	.0300	.0210	.0162	.0129	
24.	.0417	.0339	.0280	.0271	.0188	.0145	.0114	
26.	.0385	.0312	.0257	.0248	.0170	.0130	.0102	
28.	.0357	.0288	.0236	.0228	.0155	.0118	.0092	
30.	.0333	.0268	.0218	.0211	.0142	.0107	.0083	

ratio of their abscissas v to v_1 . As stated in Table II, if we wish to obtain the value of the ordinate p in linear measure we must multiply the ratios $\frac{p}{p_1}$ by p_1 expressed in linear measure. When the cut-off $\frac{AC}{HE}$ is very small we need not divide HF into four equal parts, but can then simply lay off HF as many times as possible on FE , thus getting a sufficient number of points to draw the curve CD with accuracy. The compression curve $F'E'$, Fig. 2, can be drawn in a similar manner.

Construct two indicator diagrams, one for each end of the cylinder, the initial pressure being assumed equal to 85 pounds (by gauge), the real cut-off $= \frac{1}{4}$ and the clearance $= 0.07$. Leave space on sheet for four more indicator diagrams, two of which will represent the maximum, and two the minimum, power of the engine.

I E.

DETERMINATION OF MEAN EFFECTIVE PRESSURE OF INDICATOR AND OTHER DIAGRAMS.

The problem in this case is simply to find the average height of an irregular but closed figure. This may be done in four different ways. 1st, By John Coffin's modification of the planimeter (see catalogue of Ashcroft Manuf. Co). 2d, By dividing the enclosed area by equidistant ordinates and then calculating the area by Simpson's rule (see Nystrom, p 114). 3d, By dividing the enclosed area by equidistant ordinates and changing the area included between each pair of ordinates into an equivalent rectangle and then measuring the height of the latter, the mean of all these heights will be that required. 4th, When as in indicator diagrams, the character of the bounding curves is known the average height may be calculated by means of the calculus and the results tabulated. The third method is most often used in "working up" indicator cards (diagrams), though the first is

also used when many cards are to be "worked up." Tabulated heights (or mean total pressures when clearance, back pressure and compression are neglected) are principally useful when laying out the indicator diagrams of proposed engines. The largest table of the kind with which the writer is acquainted is that prepared by Mr. Richard Buel for Du Bois' translation of Weisbach's *Mechanics*, Vol. II. (Heat Engines, etc.), pp. 479 and 482. The following Table III was taken from this work. The next Table, IV on page 18, was prepared by the writer and does not apply to all cases, but only to the case in which the cut-off is chosen with special reference to *economy of steam*. The body of the table contains the mean effective pressure p_m of indicator diagrams for different amounts of clearance and real cut-offs ϵ , or corresponding initial pressures p_1 , prescribed by Mr. Emery's formula. The back pressure was assumed = 16 pounds and both expansion and compression curves were assumed to follow the law $pv^{1.4} = \text{constant}$. Compression was supposed to be continued till the final pressure =

$$\frac{p_1 + 16}{2} = p_1 - \frac{F_o}{A} \quad (9)$$

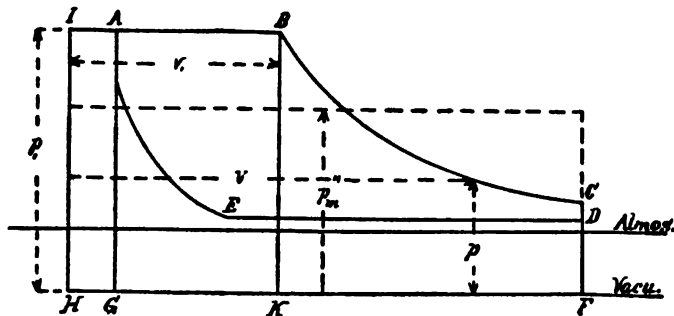


Fig. 3.

$\frac{IB}{HF}$ = real cut-off. $\frac{AB}{GF}$ = apparent cut-off. IA = clearance.
 $IH = BK$ = absolute initial pressure p_1 . $p'_m = IBCFHI \div HF$.

TABLE III.

TABLE CONTAINING VALUES OF THE RATIO $\frac{p''}{p'}$
 FOR THE STEAM CURVES pv AND $pv^{1/4}$.

Real cut-off.	pv = constant. Equilateral hyperbola.	$pv^{1/4}$ = constant. Saturation curve.	Real cut-off.	pv = constant. Equilateral hyperbola.	$pv^{1/4}$ = constant. Saturation curve.	Real cut-off.	pv = constant. Equilateral hyperbola.	$pv^{1/4}$ = constant. Saturation curve.	Real cut-off.	pv = constant. Equilateral hyperbola.	$pv^{1/4}$ = constant. Saturation curve.	Real cut-off.	pv = constant. Equilateral hyperbola.	$pv^{1/4}$ = constant. Saturation curve.
.01	.0561	.0500	.26	.6102	.5959	.51	.8534	.8463	.76	.9686	.9668			
.02	.0982	.0884	.27	.6235	.6094	.52	.8600	.8532	.77	.9713	.9696			
.03	.1352	.1245	.28	.6364	.6226	.53	.8665	.8599	.78	.9738	.9723			
.04	.1688	.1566	.29	.6490	.6355	.54	.8727	.8664	.79	.9762	.9749			
.05	.1998	.1866	.30	.6612	.6479	.55	.8788	.8727	.80	.9785	.9773			
.06	.2288	.2148	.31	.6731	.6601	.56	.8847	.8789	.81	.9807	.9796			
.07	.2562	.2415	.32	.6846	.6719	.57	.8904	.8848	.82	.9827	.9817			
.08	.2821	.2669	.33	.6959	.6835	.58	.8959	.8906	.83	.9847	.9838			
.09	.3067	.2912	.34	.7068	.6947	.59	.9013	.8962	.84	.9865	.9857			
.10	.3303	.3145	.35	.7174	.7056	.60	.9065	.9017	.85	.9881	.9874			
.11	.3528	.3368	.36	.7278	.7163	.61	.9115	.9069	.86	.9897	.9891			
.12	.3744	.3583	.37	.7379	.7267	.62	.9164	.9120	.87	.9912	.9906			
.13	.3952	.3790	.38	.7477	.7368	.63	.9211	.9169	.88	.9925	.9920			
.14	.4153	.3990	.39	.7572	.7466	.64	.9256	.9217	.89	.9937	.9933			
.15	.4346	.4183	.40	.7665	.7562	.65	.9300	.9263	.90	.9948	.9945			
.16	.4532	.4370	.41	.7756	.7656	.66	.9342	.9307	.91	.9958	.9956			
.17	.4712	.4552	.42	.7844	.7747	.67	.9383	.9350	.92	.9967	.9965			
.18	.4887	.4727	.43	.7929	.7835	.68	.9423	.9391	.93	.9975	.9973			
.19	.5055	.4897	.44	.8012	.7921	.69	.9460	.9431	.94	.9982	.9981			
.20	.5219	.5062	.45	.8093	.8005	.70	.9497	.9469	.95	.9987	.9987			
.21	.5377	.5223	.46	.8172	.8087	.71	.9532	.9506	.96	.9992	.9991			
.22	.5531	.5378	.47	.8249	.8166	.72	.9565	.9541	.97	.9996	.9995			
.23	.5680	.5530	.48	.8323	.8244	.73	.9597	.9575	.98	.9998	.9998			
.24	.5825	.5677	.49	.8395	.8319	.74	.9628	.9607	.99	.9999	.9999			
.25	.5966	.5820	.50	.8466	.8392	.75	.9658	.9638						

TABLE IV.

VALUES OF THE MEAN EFFECTIVE PRESSURES p_m

CORRESPONDING TO ECONOMICAL POINT OF CUT-OFF,

$$\epsilon = \frac{22}{22 + p_1}$$

COMPRESSION TAKES PLACE UP TO A PRESSURE $= \frac{p_1 + 16}{2} = p_1 - \frac{F_o}{A}$.

HERE ALL STEAM CURVES FOLLOW THE LAW $p_1 v_1^{1.4} = \text{CONSTANT}$.

See reference to this Table on page 16.

Real Cut-off $r = \frac{1}{r}$	Absolute pressures p_1	Clearance.				
		.01	.03	.05	.07	.09
.10	198.	44.27	34.31	26.43	18.53	10.65
.11	178.	40.62	33.94	27.27	20.61	13.96
.12	161.3	38.93	33.18	26.56	21.67	15.99
.13	147.2	37.3	32.16	27.37	22.39	17.4
.14	135.1	35.8	31.4	27.1	22.7	18.3
.15	124.7	34.3	30.4	26.6	22.8	19.0
.16	115.5	32.8	29.4	26.1	22.7	19.4
.17	107.4	31.4	28.4	25.5	22.5	19.5
.18	100.2	30.2	27.4	24.8	22.1	19.5
.19	93.8	28.8	26.4	24.0	21.7	19.3
.20	88.0	27.5	25.4	23.3	21.2	19.1
.21	82.8	26.3	24.4	22.5	20.6	18.8
.22	78.0	25.1	23.1	21.7	20.0	18.4
.23	73.7	24.0	22.5	20.9	19.4	17.9
.24	69.7	22.9	21.5	20.0	18.8	17.6
.25	66.0	21.8	20.6	19.3	18.1	16.9
.26	62.6	20.8	19.7	18.5	17.4	16.3
.27	59.5	19.8	18.7	17.7	16.7	15.7
.28	56.6	18.8	17.9	17.0	16.1	15.1
.29	53.9	17.8	17.0	16.2	15.4	14.6
.30	51.3	16.9	16.1	15.4	15.2	13.9

I F.

To determine the cut-off corresponding to a given horse-power H_1 , when the mean effective pressure p_m and the mean total pressure p''_m (see Table III and Fig. 3) is known for another horse-power H of the same engine running at the same speed, with the same initial pressure p_1 and the same back pressure and compression.

In engines with cut-off under the control of the governor, the speed (*i. e.* $R.p.m.$) remains constant. Hence we have

$$p_m : p_{m1} :: H : H_1 \text{ and } p_{m1} = \frac{H_1}{H} p_m. \quad (10)$$

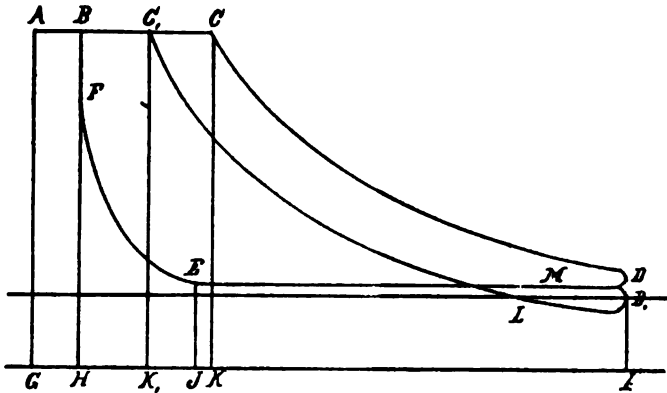


Fig. 4.

By means of Fig. 4, if we suppose the line of counter-pressures DEF to remain the same for all variations of power, we can deduce the following relations:

$$\frac{\text{area } BCDEFB}{\text{stroke } HI} = p_m \quad \frac{\text{area } BC_1LD_1MEFB}{\text{stroke } HI} = p_{m1} \quad (11)$$

$$\frac{\text{area } ACDIGA}{\text{length } GI} = p''_m \quad \frac{\text{area } AC_1LD_1JGA}{\text{length } GI} = p''_{m1} \quad (12)$$

In the figure, $\frac{AB}{HI}$ = clearance i and AG = initial pressure p_1 .

Table III on page 17 contains the values of $\frac{p''_m}{p_1}$ for various

real cut-offs and for the two steam curves $pv = \text{constant}$ and $p_1 v_1^{\frac{1}{n}} = \text{constant}$.

$$p_4 = \frac{\text{area } ABFEMDIGA}{GI} \text{ or}$$

$$p_4 = \frac{\text{area } ACDIGA - \text{area } BCDMEFB}{GH + HI} = p''_m - \frac{p_m}{1+i} \quad (13)$$

$$p''_{m1} \times GI = p_{m1} \times HI + p_4 \times GI$$

$$p''_{m1} = \frac{p_{m1}}{1+i} + p''_m - \frac{p_m}{1+i}$$

$$\text{hence } \frac{p''_{m1}}{p_1} = \frac{p_{m1} - p_m}{(1+i)p_1} + \frac{p''_m}{p_1} = \frac{p''_m}{p_1} + \left(\frac{\frac{H_1}{H} - 1}{1+i} \right) \frac{p_m}{p_1} \quad (14)$$

In a like manner it may be shown that

$$\frac{p''_{m2}}{p_1} = \frac{p''_m}{p_1} + \left(\frac{\frac{H_2}{H} - 1}{1+i} \right) \frac{p_m}{p_1}$$

where p''_{m2} is the corresponding mean total pressure for the horse power H_2 .

The quantities in the second members of these equations are either given directly or may be found from the diagrams already constructed or from the tables. This enables us to calculate

$\frac{p''_{m1}}{p_1}$ and $\frac{p''_{m2}}{p_1}$ and from these, by means of Table III on page

17, we can get the corresponding cut-offs $\frac{1}{r_1}$ and $\frac{1}{r_2}$.

Calculate the cut-offs corresponding to 50 and 170 *I. H. P.*, the extremes of power for the proposed engine. Then draw two indicator diagrams for each of these powers.

I G.

DIAGRAMS OF EFFECTIVE STEAM PRESSURES.

Draw diagram of effective steam pressures on piston both for the forward and return strokes. We will here state what is to be understood by the forward and the return stroke. In all cases suppose the engine to lie horizontally, with the crank on the right hand and the cylinder on the left. The forward stroke will then be when the piston moves to the right or when the crank moves in its upper or lower semi-circle so that it first passes through the quadrant nearest the cylinder and then through the quadrant farthest from the cylinder. This does not agree with locomotive practice where the forward end of the cylinder is its front end, and the stroke of the piston toward that end the forward stroke, but it agrees with the practice of many engineers when treating of stationary engines. To avoid all ambiguity as

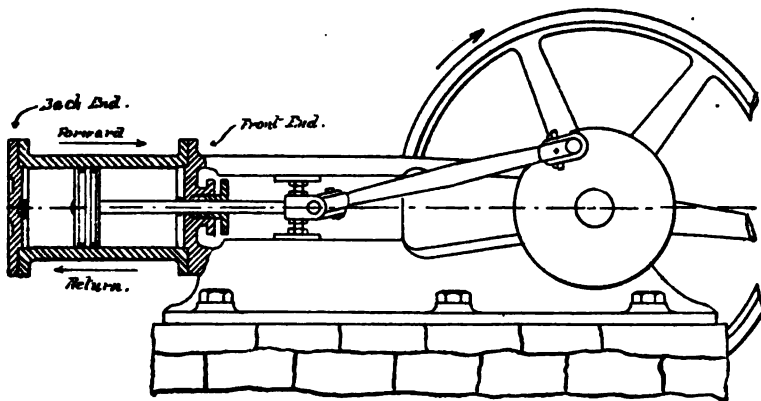


Fig. 5.

to which end of the cylinder is meant, it would be well when figures are not given to speak of the end in question as nearest to, or farthest from, the crank. In like manner we may speak of the stroke *toward* or *away from* the crank shaft, which will correspond respectively to forward and return strokes as defined above.

Fig. 6 and Fig. 7 represent respectively the indicator diagrams from back and front ends of the cylinder shown in Fig. 5. If we wish to ascertain only the power of an engine these indicator diagrams will be sufficient, for from them we can obtain in the usual manner, the power developed by the steam in each end of the cylinder, but if we wish to ascertain the real pressure

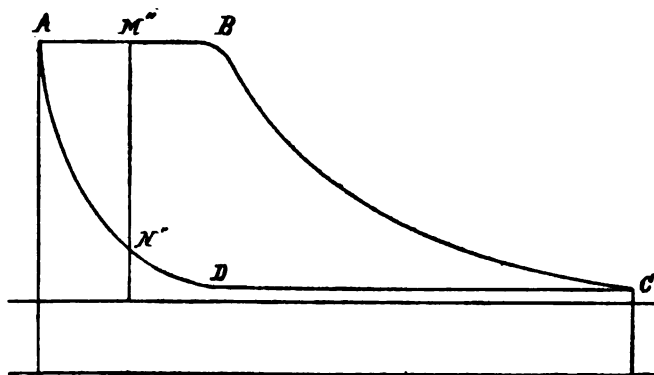


Fig. 6.

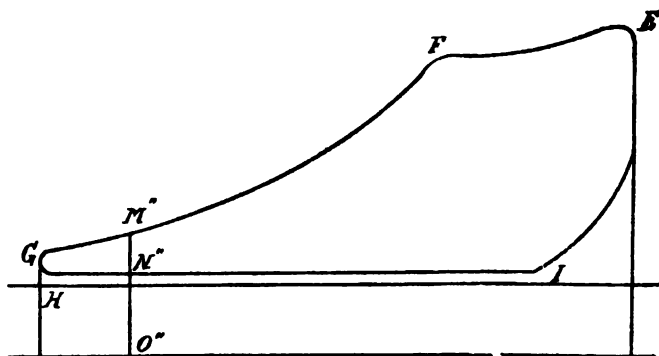


Fig. 7.

exerted by the steam upon the piston at any position of the latter we must combine both diagrams as follows.

The admission and expansion line of one end of the cylinder being placed upon the exhaust and compression line of the other end, Figs. 8 and 9, the portions of the vertical ordinates included

by the shaded portion of these figures will represent the difference between the absolute pressures of the steam on opposite sides of the piston and consequently will represent the effective pressure or driving effort exerted by the steam upon the piston for the various positions of the latter. For instance, for the piston position in Fig. 5, on the forward stroke, the effective or driving

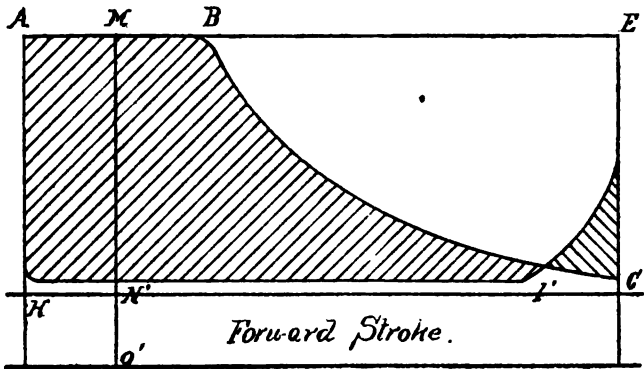


Fig. 8.

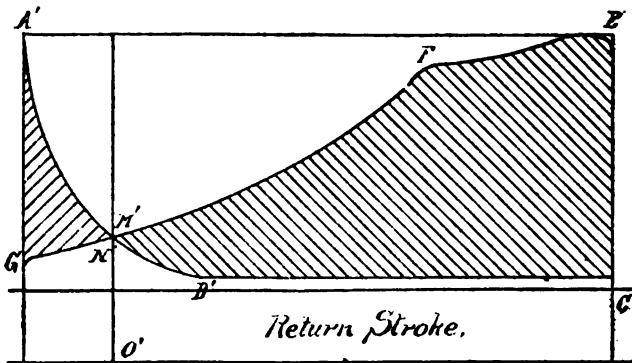


Fig. 9.

pressure on the piston will be $M^1N^1 = M^1O^1 - N^1O^1$, Fig. 8, M^1O^1 being the absolute pressure of steam in the back end of cylinder, and N^1O^1 the absolute pressure in the front end. The intercept $M^{12}N^{12}$, Fig. 6 or 7, corresponding to the same piston position is considerably different from M^1N^1 , Fig. 8 or 9, and does

not represent the difference of two pressures existing simultaneously, in opposite ends of the cylinder, but represents the difference between two pressures separated by an interval of time corresponding to about half a revolution of the crank. The effective driving pressure on the piston for the position in Fig. 5, on the return stroke is equal to zero, as is shown at M^1N^1 , Fig. 9, the pressure on opposite sides of piston being equal at that moment. The point where the lines cross represents, therefore, equilibrium between the opposing pressures; after this point is passed the resistances are in excess, and if it were not for the inertia of the reciprocating parts and fly wheel, the motion of the piston would be reversed before it had completed its stroke. This change in the direction of the excess of pressure is represented in the diagram by a difference in shading. This reversal of pressures serves a most useful purpose in absorbing the inertia of the reciprocating parts, bringing them gradually to rest and thus preventing shocks and vibrations at the end of the stroke. Draw in accordance with the above, three pairs of diagrams of effective steam pressures on piston, corresponding to 50, 110 and 170 *IHP*, the minimum, normal and maximum horse-powers of this engine.

II.

DETERMINATION OF RATIO OF LENGTH OF CONNECTING ROD TO LENGTH OF CRANK.

When the connecting rod is infinite in length (which is the case in the mechanism known as the normal double slider crank chain, Reuleaux's *Kinematics*, p. 314) the speed of the piston is the same for the crank positions ω and $180^\circ - \omega$; the distance from the nearest end of the stroke is also the same for these two crank positions, for the piston will travel through the first and second halves of its stroke while the crank pin is sweeping through the first and second quadrants of its motion. The horizontal motion of the piston along the axis of the cylinder corres-

ponds exactly to the horizontal motion of the crank pin at every instant. Moreover, the pressure on slides at right angles to piston motion = 0, the obliquity of the connecting rod being 0.

But when the connecting rod is of finite length, the piston speed and distances from nearest end of stroke are different for the crank angles ω and $180^\circ - \omega$, the paths traversed by piston while the crank is swinging through the first and second quadrants are unequal, and the pressure on the slide and consequently the friction on the slide is proportional to the ratio of the length of crank to connecting rod. In the following figure

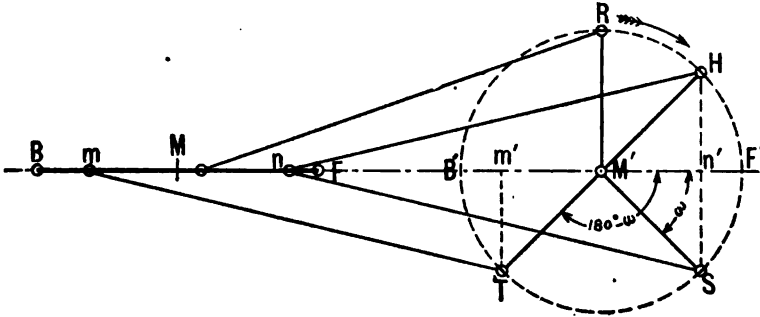


Fig. 10.

we have taken $\frac{\text{connecting rod}}{\text{crank}} = 2\frac{1}{2}$ to show the effect of a very short rod on piston speeds and positions.

If we suppose the motion of the crank pin uniform and equal to v we shall have the piston speed $w = v \sin \omega$ for each of the crank positions S and T when the connecting rod is infinite in length, but when it is finite and equal to $2\frac{1}{2}$ times the crank, as in the figure, the velocity of piston for crank positions S and T will be respectively (see Fig. 10 for ω)

$$w = v \sin \omega \left(1 + \frac{\cos \omega}{2.5} \right) \text{ for point } T$$

$$\text{and } w = v \sin \omega \left(1 - \frac{\cos \omega}{2.5} \right) \text{ for point } S. \quad (16)$$

TABLE V.

DISTANCE OF PISTON FROM BEGINNING OF STROKE WHEN STROKE IS
towards CRANK-SHAFT.

To find Actual Distance of Piston from end of Stroke, multiply tabular quantity by Stroke.

Crank Angles.	Connecting-rod ÷ Crank ==								Crank Angles.	Connecting-rod ÷ Crank ==							
	4	4.5	5	5.5	6	7	8	∞		4	4.5	5	5.5	6	7	8	∞
2	.0034	.0004	.0004	.0004	.0004	.0003	.0003	.0003	92	.5800	.5737	.5679	.5632	.5594	.5533	.5487	.5174
4	.0015	.0015	.0015	.0014	.0014	.0014	.0014	.0012	94	.5981	.5909	.5851	.5805	.5768	.5706	.5661	.5349
6	.0034	.0033	.0033	.0032	.0032	.0031	.0030	.0027	96	.6151	.6079	.6022	.5977	.5938	.5878	.5833	.5523
8	.0061	.0060	.0058	.0057	.0057	.0056	.0055	.0049	98	.6318	.6247	.6191	.6145	.6107	.6048	.6004	.5694
10	.0095	.0092	.0091	.0089	.0089	.0087	.0085	.0070	100	.6484	.6414	.6358	.6313	.6275	.6216	.6172	.5868
12	.0136	.0133	.0131	.0129	.0127	.0126	.0124	.0109	102	.6647	.6578	.6523	.6478	.6441	.6384	.6340	.6040
14	.0185	.0181	.0178	.0175	.0173	.0171	.0170	.0149	104	.6807	.6739	.6685	.6641	.6605	.6548	.6505	.6218
16	.0241	.0236	.0232	.0228	.0225	.0221	.0218	.0194	106	.6964	.6897	.6845	.6802	.6766	.6710	.6668	.6381
18	.0304	.0298	.0293	.0289	.0285	.0279	.0275	.0245	108	.7110	.7053	.7002	.6959	.6924	.6870	.6829	.6545
20	.0375	.0367	.0360	.0355	.0350	.0344	.0339	.0302	110	.7270	.7203	.7155	.7115	.7080	.7027	.6987	.6710
22	.0452	.0442	.0434	.0428	.0423	.0414	.0408	.0364	112	.7418	.7356	.7307	.7267	.7233	.7181	.7143	.6873
24	.0536	.0524	.0515	.0507	.0501	.0491	.0484	.0432	114	.7562	.7502	.7455	.7416	.7383	.7334	.7296	.7034
26	.0627	.0614	.0604	.0594	.0586	.0575	.0568	.0506	116	.7703	.7645	.7599	.7561	.7529	.7482	.7445	.7194
28	.0724	.0708	.0696	.0686	.0677	.0664	.0654	.0585	118	.7841	.7785	.7740	.7704	.7674	.7626	.7591	.7347
30	.0827	.0809	.0796	.0784	.0774	.0759	.0748	.0670	120	.7974	.7921	.7878	.7843	.7814	.7769	.7735	.7500
32	.0936	.0916	.0901	.0888	.0877	.0860	.0848	.0760	122	.8104	.8053	.8012	.7979	.7951	.7908	.7875	.7650
34	.1051	.1029	.1012	.0997	.0985	.0967	.0953	.0855	124	.8230	.8181	.8142	.8105	.8084	.8042	.8011	.7796
36	.1174	.1147	.1128	.1112	.1099	.1079	.1063	.0955	126	.8352	.8305	.8268	.8238	.8213	.8173	.8144	.7939
38	.1298	.1271	.1250	.1233	.1218	.1196	.1179	.1060	128	.8470	.8426	.8391	.8362	.8338	.8300	.8273	.8078
40	.1430	.1401	.1377	.1358	.1342	.1318	.1297	.1170	130	.8584	.8542	.8509	.8482	.8459	.8424	.8398	.8214
42	.1566	.1534	.1509	.1489	.1471	.1444	.1424	.1284	132	.8694	.8655	.8623	.8598	.8577	.8544	.8519	.8346
44	.1707	.1673	.1646	.1623	.1605	.1575	.1557	.1403	134	.8799	.8763	.8733	.8710	.8689	.8658	.8635	.8477
46	.1853	.1817	.1787	.1763	.1743	.1712	.1689	.1527	136	.8901	.8866	.8839	.8817	.8796	.8770	.8751	.8597
48	.2005	.1965	.1922	.1906	.1885	.1852	.1827	.1654	138	.8998	.8966	.8941	.8920	.8903	.8878	.8856	.8710
50	.2153	.2113	.2081	.2064	.2032	.1996	.1967	.1786	140	.9090	.9061	.9038	.9019	.9003	.8978	.8959	.8836
52	.2314	.2270	.2234	.2206	.2182	.2144	.2117	.1922	142	.9178	.9152	.9130	.9113	.9098	.9076	.9059	.8940
54	.2474	.2428	.2391	.2360	.2335	.2293	.2266	.2061	144	.9262	.9238	.9218	.9203	.9189	.9169	.9153	.9045
56	.2638	.2589	.2550	.2518	.2492	.2451	.2420	.2204	146	.9342	.9320	.9302	.9288	.9276	.9257	.9243	.9145
58	.2805	.2754	.2713	.2679	.2652	.2608	.2575	.2350	148	.9417	.9397	.9381	.9368	.9355	.9334	.9328	.9240
60	.2974	.2920	.2878	.2843	.2814	.2769	.2735	.2501	150	.9487	.9469	.9455	.9444	.9438	.9419	.9408	.9330
62	.3146	.3090	.3046	.3009	.2979	.2932	.2897	.2653	152	.9553	.9538	.9525	.9515	.9507	.9494	.9484	.9415
64	.3320	.3262	.3215	.3178	.3147	.3098	.3061	.2808	154	.9617	.9601	.9590	.9581	.9574	.9563	.9554	.9494
66	.3495	.3435	.3387	.3348	.3316	.3266	.3228	.2966	156	.9671	.9660	.9651	.9643	.9637	.9627	.9620	.9568
68	.3672	.3610	.3561	.3520	.3487	.3435	.3397	.3127	158	.9724	.9713	.9706	.9700	.9694	.9686	.9678	.9634
70	.3850	.3786	.3735	.3694	.3660	.3607	.3567	.3290	160	.9772	.9763	.9757	.9751	.9747	.9740	.9735	.9694
72	.4028	.3963	.3911	.3869	.3834	.3780	.3739	.3455	162	.9815	.9808	.9803	.9801	.9795	.9789	.9785	.9755
74	.4208	.4141	.4088	.4045	.4009	.3954	.3912	.3622	164	.9854	.9848	.9844	.9841	.9838	.9834	.9830	.9806
76	.4388	.4319	.4266	.4222	.4185	.4128	.4085	.3790	166	.9888	.9884	.9881	.9878	.9878	.9872	.9869	.9851
78	.4568	.4498	.4444	.4399	.4362	.4304	.4260	.3960	168	.9918	.9913	.9912	.9910	.9909	.9907	.9905	.9891
80	.4747	.4677	.4622	.4576	.4539	.4480	.4436	.4132	170	.9943	.9941	.9939	.9938	.9937	.9935	.9933	.9924
82	.4927	.4857	.4799	.4754	.4716	.4656	.4612	.4304	172	.9963	.9962	.9961	.9960	.9959	.9958	.9957	.9951
84	.5105	.5034	.4977	.4930	.4892	.4832	.4787	.4477	174	.9979	.9979	.9978	.9978	.9977	.9977	.9976	.9973
86	.5283	.5211	.5154	.5107	.5069	.5008	.4963	.4651	176	.9990	.9990	.9990	.9990	.9990	.9990	.9990	.9988
88	.5460	.5388	.5330	.5283	.5245	.5185	.5139	.4826	178	.9998	.9997	.9997	.9997	.9997	.9997	.9997	.9997
90	.5535	.5563	.5505	.5453	.5400	.5339	.5294	.4980	180	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000

DISTANCE OF PISTON FROM BEGINNING OF STROKE WHEN STROKE IS
away from CRANK-SHAFT.

Crank Angles.	Connecting-rod ÷ Crank =								Crank Angles.	Connecting-rod ÷ Crank =							
	4	4.5	5	5.5	6	7	8	∞		4	4.5	5	5.5	6	7	8	∞
2	.0002	.0003	.0003	.0003	.0003	.0003	.0003	.0003	92	.4540	.4612	.4670	.4717	.4755	.4715	.4861	.5174
4	.0010	.0010	.0010	.0010	.0010	.0010	.0010	.0012	94	.4717	.4789	.4846	.4893	.4931	.4932	.5037	.5349
6	.0021	.0021	.0022	.0022	.0023	.0023	.0024	.0027	96	.4895	.4966	.5023	.5070	.5108	.5168	.5213	.5523
8	.0036	.0038	.0039	.0040	.0041	.0042	.0043	.0049	98	.5073	.5143	.5201	.5246	.5284	.5344	.5388	.5699
10	.0057	.0059	.0061	.0062	.0063	.0065	.0067	.0076	100	.5253	.5323	.5378	.5424	.5461	.5520	.5564	.5868
12	.0082	.0087	.0088	.0090	.0091	.0093	.0095	.0109	102	.5432	.5502	.5556	.5601	.5638	.5696	.5740	.6140
14	.0112	.0116	.0119	.0122	.0122	.0128	.0131	.0149	104	.5612	.5681	.5734	.5778	.5815	.5872	.5915	.6316
16	.0146	.0152	.0156	.0159	.0162	.0167	.0170	.0194	106	.5792	.5859	.5912	.5955	.5991	.6046	.6088	.6489
18	.0185	.0192	.0197	.0199	.0205	.0211	.0215	.0245	108	.5972	.6037	.6089	.6131	.6166	.6220	.6261	.6715
20	.0228	.0237	.0243	.0248	.0253	.0260	.0265	.0302	110	.6150	.6214	.6265	.6306	.6340	.6393	.6533	.6940
22	.0276	.0286	.0294	.0300	.0306	.0314	.0322	.0366	112	.6328	.6390	.6439	.6480	.6513	.6565	.6603	.6873
24	.0329	.0340	.0349	.0357	.0363	.0373	.0380	.0432	114	.6505	.6565	.6613	.6652	.6684	.6734	.6772	.7034
26	.0385	.0399	.0410	.0419	.0426	.0437	.0446	.0506	116	.6680	.6738	.6785	.6822	.6853	.6902	.6939	.7192
28	.0447	.0462	.0475	.0485	.0493	.0505	.0516	.0585	118	.6844	.6910	.6954	.6991	.7021	.7068	.7103	.7347
30	.0513	.0531	.0545	.0555	.0565	.0581	.0592	.0670	120	.7026	.7080	.7122	.7157	.7186	.7231	.7265	.7500
32	.0583	.0603	.0619	.0632	.0642	.0660	.0671	.0760	122	.7195	.7245	.7287	.7321	.7348	.7392	.7425	.7650
34	.0658	.0680	.0698	.0712	.0724	.0747	.0757	.0855	124	.7362	.7411	.7450	.7482	.7508	.7550	.7581	.7796
36	.0738	.0762	.0782	.0797	.0811	.0838	.0847	.0955	126	.7526	.7572	.7609	.7640	.7665	.7705	.7734	.7939
38	.0822	.0848	.0870	.0887	.0902	.0924	.0941	.1060	128	.7686	.7730	.7766	.7794	.7818	.7856	.7883	.8078
40	.0910	.0939	.0962	.0981	.0997	.1022	.1041	.1170	130	.7844	.7885	.7919	.7949	.7969	.8004	.8030	.8214
42	.1002	.1034	.1059	.1080	.1097	.1124	.1144	.1284	132	.7997	.8037	.8068	.8094	.8115	.8148	.8173	.8346
44	.1099	.1134	.1161	.1183	.1202	.1230	.1249	.1403	134	.8147	.8184	.8213	.8237	.8257	.8288	.8311	.8473
46	.1201	.1237	.1267	.1290	.1310	.1342	.1365	.1527	136	.8293	.8327	.8354	.8377	.8395	.8424	.8443	.8597
48	.1306	.1348	.1377	.1402	.1423	.1456	.1481	.1654	138	.8434	.8469	.8491	.8511	.8529	.8556		

With respect to distance of piston from end of stroke it is evident from the figure that the distances mB and nF , corresponding to the crank positions T and S , are unequal. If the connecting rod were infinite $m'B^1$ and $n'F^1$ would respectively represent the distances of piston from nearest end of stroke for the crank positions T and S .

The following figure, drawn to scale, shows graphically the inequalities of the paths traversed by piston during the first and

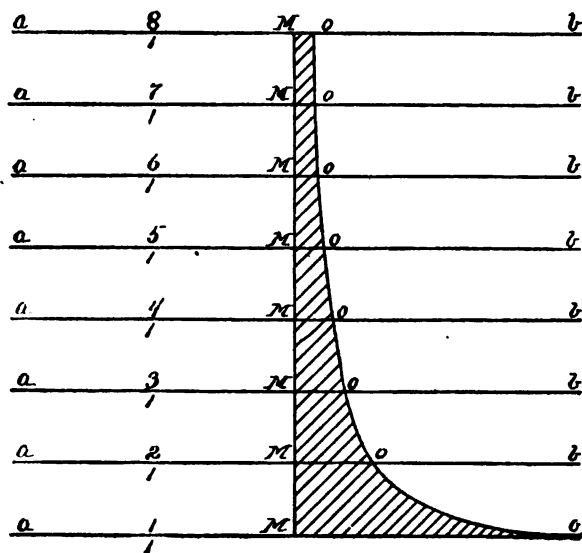


Fig. 11.

second quadrants of crank motion for different ratios of connecting rod to crank, the horizontal position of stroke ($= ab$) included by shaded figure representing the distance Mo of the piston from its middle position M (see preceding figure and Tables V and VI).

When length of crank $= R = 1$ and $\omega = 90^\circ$

$\frac{L}{R} =$	8	7	6	5	4	3	2	1
$Mo. =$	.063	.072	.084	.101	.127	.172	.268	1.0

Fig. 11 shows clearly that as the ratio of connecting rod to crank diminishes the inequalities of the paths ao and ob respec-

tively described by piston while crank passes through the first and second quadrants, increases. This inequality tends to heap up the work in the first and fourth quadrants, as will now be more fully shown.

In order to show how the length of connecting rod influences the tangential components of the pressures on crank pin, we will assume the horizontal pressure on cross-head pin constant, and

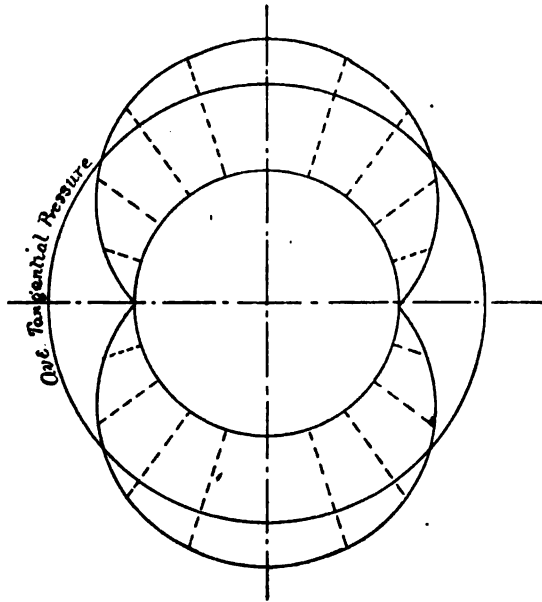


Fig. 12.

will lay off the tangential pressures on prolongation of the radii of the crank-pin circle, Figs. 12 and 13. The curve drawn through the extremities of these radial ordinates will show the variation of the tangential pressure for the different crank positions; the area included between this curve and the crank pin circle measures, approximately, the work done in one revolution. The two diagrams drawn correspond respectively to $\frac{\text{connecting rod}}{\text{crank}} = \text{infinity}$ and $\frac{\text{connecting rod}}{\text{crank}} = 2\frac{1}{2}$ as above. A glance at these diagrams shows

at once that while with the infinite rod the work done in each of the quadrants is equal, with the finite rod the principal part of the work is done in quadrants I and IV. The average tangential pressures can easily be found by remembering that if friction is neglected the work done on piston must equal the work done on crank pin, and that therefore the average tangential pressure p_t ,

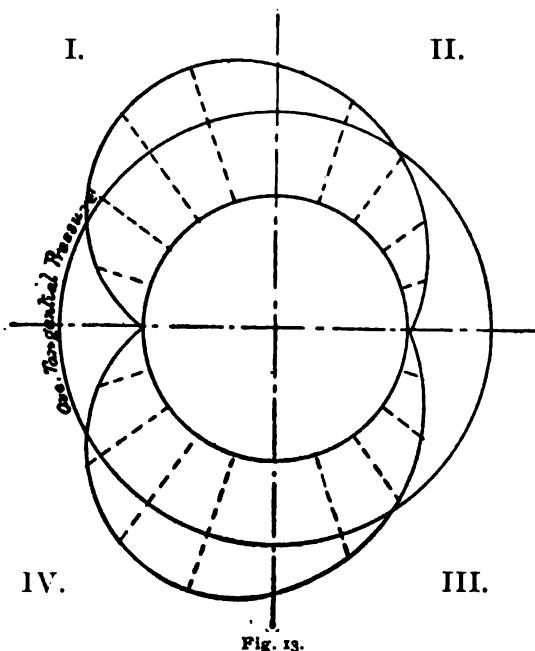


Fig. 13.

must be to the average piston pressure p_m as the path $2S$ (S = stroke) of piston in one revolution is to the path πS of the crank pin, or

$$p_t : p_m :: 2S : \pi S \quad \text{hence} \quad p_t = \frac{2}{\pi} p_m = 0.6366 p_m. \quad (17)$$

Attention has already been called to the fact that the pressure and friction on the slide increase with the ratio of crank to connecting rod, then the

$$\text{max. friction on slides} = \mu P \times \frac{\text{crank}}{\text{connecting rod}} \quad (18)$$

μ = coefficient of friction, P = pressure on cross-head pin.

But the ratio of crank to connecting rod affects not only the distribution of work and the magnitude of friction, it also affects the distribution of steam by increasing the duration of admission and reducing that of exhaust at one end of the cylinder, while it diminishes the period of admission and increases the duration of exhaust at the other end.

That it thus affects the admission of steam can be roughly shown by means of Fig. 10, page 25. Since the ratio of eccentric radius to length of eccentric rod is small, even when the ratio

$\frac{\text{crank}}{\text{connecting rod}}$ is great, we may neglect the obliquity of eccentric rod and treat the latter as if it were of infinite length, consequently the angular positions of the eccentric radius for the points of cut-off will be diametrically opposite, and since eccentric and crank are keyed to the same shaft, the crank positions corresponding to points of cut-off will also be directly opposite. Let H and T , Fig. 10, page 25, represent the two crank positions corresponding to cut-off, it will then be evident from the figure that the period of admission Bm for the end of the cylinder farthest from the crank will be greater than the admission Fn for end of cylinder nearest crank. When the ratio of crank to connecting rod is large this distribution of steam will be injurious, but when it is equal to, or less than $\frac{1}{2}$ this irregularity of admission and exhaust at the two ends will not only not be excessive but will be a positive advantage in high speed engines, because it gives the greater compression (*i. e.* diminished period of exhaust) at the end farthest from the crank where it is most needed. This will be proved later when we come to consider the proper period of compression. Moreover, with such proportions of crank to connecting rod, the valve setting obtained from Zeuner's simple valve diagram will give very satisfactory results practically. The

indicator diagrams for the two ends of the cylinder obtained from such distribution of steam will of course not be identical in form, but they will be nearly equal in area, and as we have already mentioned will give the greater compression where it is most needed. In marine engines where engine space is limited the ratio is sometimes as low as $1\frac{1}{2}$, while in other well designed stationary engines it is sometimes as high as 8. Decide upon the proper ratio for the present case.

III.

DETERMINATION OF THAT MEAN ACCELERATING FORCE (PER $\square''$
OF PISTON) NECESSARY TO START RECIPROCATING PARTS,
WHICH CORRESPONDS TO THE MOST UNIFORM
TANGENTIAL PRESSURE ON CRANK PIN.

That the weight and speed of the reciprocating parts have an important influence on the steadiness with which an engine can be run, will be evident from the following consideration: Steadiness of running, other things being equal, results from the driving effort, or tangential pressure on crank pin, being as nearly constant as possible for all positions of the crank; the magnitude of the tangential pressure ($=DF$ in Fig. 14) depends upon the position of the crank CD , and upon the direction and intensity of the pressure $DB' = EB$ exerted by the connecting rod upon the crank pin; this latter pressure in turn depends upon the angular position of the connecting rod and upon the magnitude of the force $K = EA$. Let us now suppose the mass M of the reciprocating parts (piston, piston rod, cross head and connecting rod)* to be concentrated in a ring around the cross head

* According to Weisbach-Herrmann, (Vol. III, Section II, p. 867) the work stored up in a connecting rod of uniform section is the same as if one-third of its mass were concentrated at crank pin and the remaining two-thirds at cross-head pin. This result is itself an approximation and as sections are generally not uniform the accuracy of the assumption is still further diminished. Usually rod tapers from wrist to crank-pin.

pin, and that the resistance offered by this mass to a change in its state of motion, is represented by F , we will then have $F = M\theta$ where θ represents the acceleration of the center of the cross head pin, M = mass of reciprocating parts and F = force necessary to give the mass M the acceleration θ .

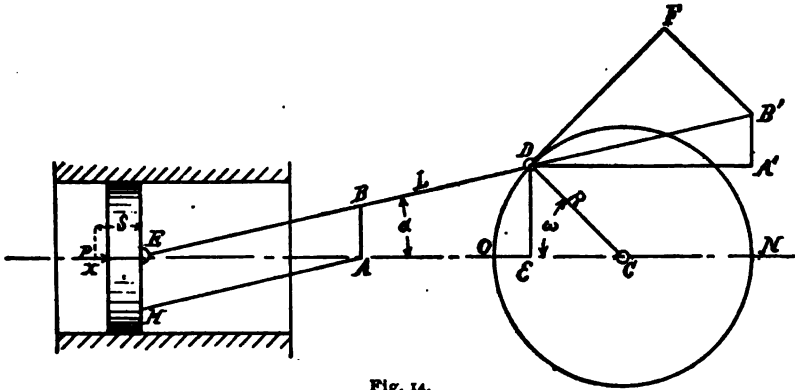


Fig. 14.

The accelerating force F is of course part of the steam pressure P , the difference between them, $P - F = K = EA$ (see figure), is that part of the steam pressure which is transmitted through the piston rod to the lower end of connecting rod and from these to the crank pin. When the point E (*i. e.* the piston) has reached its maximum velocity, acceleration ceases, and we have $F = 0$ and $P = K = EA = DA'$, but retardation now begins (for the reciprocating parts must be brought to rest at the end of the stroke) and no part of the steam pressure being absorbed in producing acceleration, the whole of the steam pressure P is transmitted to the cross head pin. This P is however not the only pressure to which the crank pin is subjected, for the reciprocating parts having once attained their maximum velocity, tend to keep it, and tend to move faster (horizontally) than the crank pin, and will thus exert a pressure F upon the latter, over and above that received from the steam ($= P$), hence

$$P + F = K = EA = DA'.$$

The general expression for the horizontal pressure exerted upon the crank pin or cross head pin will then be

$$DA' = EA = K = P \pm F$$

the *upper* sign of F being employed when reciprocating parts are retarded, and the *lower* sign, when the reciprocating parts are accelerated, F being the accelerating or retarding force necessary to produce in the mass M the acceleration or retardation θ , that is,

$$F = M\theta = \frac{W}{g}\theta \quad (19)$$

so that F increases with the weight W of the reciprocating parts and their acceleration.

Before determining the value of θ for the different piston and crank positions, we will call attention to, and emphasize the point, that the horizontal pressures on crank pin or cross head pin are not directly obtainable from the diagram of effective steam pressures on piston but can be obtained indirectly from the latter by *subtracting* the *accelerating* pressures from the steam pressures given by the diagram *for the earlier part of the stroke*, and *adding* the *retarding* pressures to the steam pressures given by the diagram *for the later part of the stroke*. From this we already see that the mass and motion of the reciprocating parts tend to equalize the horizontal pressures upon the crank pin by diminishing the larger pressures near the beginning and increasing the smaller pressures near the end of expansion, the energy of the steam absorbed by the reciprocating parts near the beginning, being given out again to the crank pin toward the end, of the expansion. With too great a speed, however, this equalizing tendency will be more than neutralized by the transfer of the whole of the steam pressures to the end of the stroke and will tend to cause shocks and dangerous stresses on all the working pieces. This point will be more fully discussed further on.

We will now find an expression for the acceleration θ and then substitute in

$$F = \frac{W}{g}\theta. \quad (19)$$

The distance s traversed by cross-head-pin E from the beginning X of its stroke (see Fig. 14) is equal to

$$s = XE = XC - EC = (R + L) - (L \cos \alpha + R \cos \omega)$$

$$s = R(1 - \cos \omega) + L(1 - \cos \alpha). \quad (20)$$

This equation holds for both strokes, provided we estimate ω from 0° to 360° and s from the same dead point (left hand dead point X , Fig. 14). But if we estimate ω and s from the opposite center or dead point for the return stroke, and this is commonly done, we must use

$$s = R(1 - \cos \omega) - L(1 - \cos \alpha).$$

The cosine of the angle made by connecting rod with axis of cylinder can be obtained as follows

$$D\mathfrak{S} = L \sin \alpha = R \sin \omega$$

hence
$$\sin \alpha = \frac{R}{L} \sin \omega \quad (21)$$

and
$$\cos \alpha = \sqrt{1 - \frac{R^2}{L^2} \sin^2 \omega} = 1 - \frac{1}{2} \frac{R^2}{L^2} \sin^2 \omega \text{ very nearly } (22)$$

Introducing this value of cosine α in equation (20) we obtain

$$s = R \left[(1 - \cos \omega) + \frac{1}{2} \frac{R}{L} \sin^2 \omega \right] \quad (23)$$

For the return stroke, and estimating s and ω from the right hand dead point this equation becomes

$$s = R \left[(1 - \cos \omega) - \frac{1}{2} \frac{R}{L} \sin^2 \omega \right]$$

From this general formula for the travel, s , of the piston we can obtain the velocity, v , of the piston or of the point E when the crank has a uniform velocity.

According to Elementary Mechanics we have

$$\text{for linear velocity, } v = \frac{ds}{dt}$$

$$\text{and for acceleration, } a = \frac{dv}{dt}$$

Now from (23)

$$ds = R(\sin \omega + \frac{1}{2} \frac{R}{L} \sin 2\omega) d\omega$$

therefore

$$w = R \left(\sin \omega + \frac{1}{2} \frac{R}{L} \sin 2\omega \right) \frac{d\omega}{dt} \quad (24)$$

is velocity of piston.

If the crank pin has a uniform circumferential velocity v we will have

$$Rd\omega = vdt \text{ or } \frac{d\omega}{dt} = \frac{v}{R}. \quad (25)$$

Substituting this value of $\frac{d\omega}{dt}$ in (24) we obtain

$$w = v \left(\sin \omega + \frac{1}{2} \frac{R}{L} \sin 2\omega \right); \quad (26)$$

For return stroke and estimation of s , ω and w from the right hand dead point, we have instead

$$w = v \left(\sin \omega - \frac{1}{2} \frac{R}{L} \sin 2\omega \right).^*$$

Differentiating (26) we have

$$dw = v \left(\cos \omega d\omega + \frac{1}{2} \frac{R}{L} \cos 2\omega 2d\omega \right),$$

reducing and dividing both members by dt we obtain

$$\frac{dw}{dt} = v \left(\cos \omega + \frac{R}{L} \cos 2\omega \right) \frac{d\omega}{dt}.$$

Now since the acceleration of a unit of mass is $\theta = \frac{dw}{dt}$ we can substitute θ in preceding equation, getting

$$\theta = v \left(\cos \omega + \frac{R}{L} \cos 2\omega \right) \frac{d\omega}{dt}, \quad (27)$$

* An exact expression is:

$$w = v \left[\sin \omega \pm \frac{R}{L} \frac{\sin 2\omega}{\sqrt{1 - \frac{R^2}{L^2} \sin^2 \omega}} \right]$$

For exact ratios of w to v see Table VIII, p. 60.

but we found from (25) that $\frac{d\omega}{dt} = \frac{v}{R}$, therefore,

$$\theta = \frac{v^2}{R} \left[\cos \omega + \frac{R}{L} \cos 2\omega \right]. * \quad (28)$$

For return stroke, and estimation of s , w , ω and θ from the right hand dead point, this becomes

$$\theta = \frac{v^2}{R} \left(\cos \omega - \frac{R}{L} \cos 2\omega \right).$$

At this point we should note that the factor $\frac{v^2}{R}$ is identical with the acceleration of a unit of mass rotating in the crank pin circle, and when the connecting rod L is infinite the value of θ for $\omega = 0$ and $\omega = 180^\circ$ is $\theta = \pm \frac{v^2}{R}$, that is with infinite connecting rod the acceleration of reciprocating parts, when the crank is on the dead centers O and N , is identical with that possessed by a unit of mass rotating in the crank pin circle. But when the connecting rod is of finite length, θ is no longer equal to $\frac{v^2}{R}$, either on dead center O or on N , but we have instead when $\omega = 0$

$$\theta = \frac{v^2}{R} \left(1 + \frac{R}{L} \right) \quad (29)$$

and when $\omega = 180^\circ$

$$\theta = \frac{v^2}{R} \left(1 - \frac{R}{L} \right) \quad (30)$$

* An exact expression for θ is:

$$\theta = \frac{v^2}{R} \left[\cos \omega \mp \frac{\sin^2 \omega}{\sqrt{\frac{L^2}{R^2} - \sin^2 \omega}} \pm \frac{L^2}{R^2} \frac{\cos^2 \omega}{\left(\frac{L^2}{R^2} - \sin^2 \omega \right)^{\frac{3}{2}}} \right]$$

Table VII on p. 39 was not computed from this formula, but from bracketed part of equation 28. The table to be perfectly exact should have been computed from the bracketed part of the equation just given. The difference is so slight, however, that it can be neglected in the problems arising in practice. The upper signs correspond to forward stroke.

the numerical mean of these two values is, however, equal to $\frac{v^2}{R}$; we call this mean θ_m , or

$$\theta_m = \frac{\frac{v^2}{R} \left(1 - \frac{R}{L}\right) + \frac{v^2}{R} \left(1 + \frac{R}{L}\right)}{2} = \frac{v^2}{R} \quad (31)$$

Returning now to the expression deduced above for θ and substituting in

$$F = M\theta = \frac{W}{g}\theta \quad (32)$$

we get
$$F = \frac{W}{g} \frac{v^2}{R} \left(\cos \omega + \frac{R}{L} \cos 2\omega \right). \quad (33)$$

Dividing both members of the equation by A , area of piston in square inches, we get

$$\frac{F}{A} = \frac{W}{g} \times \frac{v^2}{R} \times \frac{1}{A} \left(\cos \omega + \frac{R}{L} \cos 2\omega \right) \quad (34)$$

If the weight W of the reciprocating parts were rotating in the crank pin circle, the centrifugal force would be

$$F_o = M \frac{v^2}{R} = \frac{W}{g} \frac{v^2}{R}; \quad (35)$$

substituting this in the above expression for $\frac{F}{A}$ we get

$$\frac{F}{A} = \left(\cos \omega + \frac{R}{L} \cos 2\omega \right) \frac{F_o}{A} = c \frac{F_o}{A} \quad (36)$$

For return stroke, and estimation of s , ω , w , and θ from the right hand dead point, this becomes

$$\frac{F}{A} = \left(\cos \omega - \frac{R}{L} \cos 2\omega \right) \frac{F_o}{A} = c \frac{F_o}{A}.$$

It is evident that the value of c will be the same numerically for angle ω of the forward stroke and angle $180^\circ - \omega$ for the return stroke, but its algebraic sign will be different in these two cases.

Table VII on following page contains values of c for different crank angles ω and different values of $\frac{L}{R}$.

TABLE VII.

ACCELERATION OF RECIPROCATING PARTS $= c \frac{F}{A}$.TABLE CONTAINING VALUES OF $c = \left(\cos \omega \pm \frac{R}{L} \cos 2\omega. \right)$ The $\left\{ \begin{array}{l} \text{upper} \\ \text{lower} \end{array} \right\}$ signs in formula relate respectively to $\left\{ \begin{array}{l} \text{forward} \\ \text{return} \end{array} \right\}$ stroke.

The algebraic signs in the table relate to forward stroke only; for return stroke, signs opposite to those there given must be used.

Crank Angles.		Connecting-rod $\div$ Crank =					Crank Angles.		Connecting-rod $\div$ Crank =				
For- ward.	Return.	4.0	4.5	5.0	5.5	6.0	For- ward.	Return.	4.0	4.5	5.0	5.5	6.0
0°	180°	1.250	1.222	1.200	1.182	1.167	90°	90°	-.250	-.222	-.200	-.182	-.167
30°	177°	1.248	1.220	1.198	1.180	1.165	87°	87°	-.301	-.277	-.250	-.233	-.218
60°	174°	1.240	1.212	1.191	1.173	1.158	84°	84°	-.350	-.322	-.301	-.283	-.268
90°	171°	1.226	1.199	1.178	1.161	1.141	81°	81°	-.394	-.367	-.346	-.329	-.315
120°	168°	1.206	1.181	1.161	1.144	1.130	78°	78°	-.430	-.411	-.391	-.374	-.360
150°	165°	1.183	1.158	1.139	1.124	1.110	75°	75°	-.470	-.449	-.430	-.415	-.401
180°	162°	1.153	1.131	1.113	1.098	1.086	72°	72°	-.511	-.489	-.471	-.456	-.444
210°	159°	1.120	1.099	1.083	1.069	1.058	69°	69°	-.544	-.523	-.510	-.491	-.482
240°	156°	1.081	1.063	1.048	1.036	1.026	66°	66°	-.574	-.556	-.541	-.529	-.519
270°	153°	1.038	1.022	1.009	.998	.989	63°	63°	-.601	-.585	-.572	-.561	-.553
300°	150°	.991	.977	.966	.957	.949	60°	60°	-.625	-.611	-.600	-.591	-.583
330°	147°	.941	.929	.920	.913	.907	57°	57°	-.647	-.635	-.626	-.619	-.616
360°	144°	.886	.878	.871	.865	.861	54°	54°	-.665	-.657	-.650	-.644	-.640
390°	141°	.829	.823	.819	.815	.812	51°	51°	-.681	-.675	-.671	-.667	-.664
420°	138°	.769	.766	.764	.762	.760	48°	48°	-.695	-.692	-.689	-.688	-.686
450°	135°	.707	.707	.707	.707	.707	45°	45°	-.707	-.707	-.707	-.707	-.707
480°	132°	.643	.646	.648	.650	.652	42°	42°	-.717	-.720	-.722	-.724	-.726
510°	129°	.577	.583	.587	.591	.594	39°	39°	-.725	-.731	-.735	-.739	-.742
540°	126°	.511	.519	.526	.532	.536	36°	36°	-.732	-.740	-.747	-.753	-.758
570°	123°	.443	.455	.464	.469	.477	33°	33°	-.737	-.749	-.758	-.765	-.771
600°	120°	.375	.389	.400	.409	.417	30°	30°	-.741	-.755	-.766	-.775	-.783
630°	117°	.307	.323	.336	.347	.356	27°	27°	-.744	-.761	-.773	-.784	-.793
660°	114°	.240	.259	.273	.285	.296	24°	24°	-.747	-.765	-.780	-.792	-.803
690°	111°	.172	.193	.209	.223	.234	21°	21°	-.748	-.769	-.785	-.798	-.810
720°	108°	.107	.129	.147	.162	.174	18°	18°	-.749	-.771	-.789	-.804	-.816
750°	105°	.032	.067	.086	.102	.115	15°	15°	-.750	-.774	-.793	-.809	-.822
780°	102°	-.020	.005	.025	.032	.056	12°	12°	-.750	-.775	-.795	-.810	-.826
810°	99°	-.082	-.055	-.034	-.017	-.003	9°	9°	-.750	-.777	-.797	-.815	-.830
840°	96°	-.140	-.112	-.090	-.073	-.058	74°	74°	-.750	-.778	-.799	-.817	-.832
870°	93°	-.197	-.169	-.147	-.130	-.114	71°	71°	-.750	-.778	-.800	-.818	-.833
900°	90°	-.250	-.222	-.200	-.182	-.167	68°	68°	-.750	-.778	-.800	-.818	-.833

From the above expression for $\frac{F}{A}$ we can deduce an approximate, graphical, method sufficiently accurate for most purposes.

If we make $\omega = 90^\circ$ and 270° we get, respectively,

$$\frac{F}{A} = -\frac{R}{L} \frac{F_0}{A} \text{ and } +\frac{R}{L} \frac{F_0}{A}; \quad (37)$$

other crank angle the horizontal line included between the two circles will measure the corresponding accelerating or retarding force.

EXACT GRAPHICAL METHOD OF FINDING ACCELERATION OF
PISTON AND OF ANY POINT ON THE ROD.

The point P in Fig. 16 is the instantaneous center for the relative motion of connecting rod to engine bed. The velocity of crank pin B is to that of slide A as the instantaneous radius PB

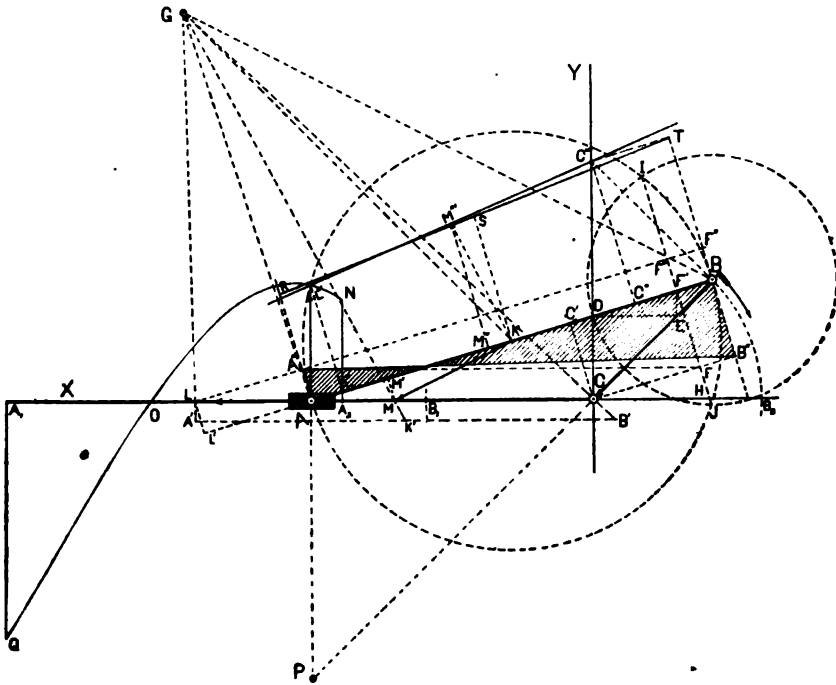


Fig. 16.

is to radius PA . As each of the radii is also at right angles to the direction of the velocity they may respectively be taken as the representatives of the velocities of points B and A . As velocities may be combined and resolved like forces, we may regard APB as a triangle of velocities and either side as the resultant of the other two. Thus PA may be considered as the resul-

tant of PB and BA . That is, the velocity of point A may be considered as the resultant of the velocity of point B about P and of the velocity of point A about B as a center (the angular velocity about B being then equal to that of B around P).*

If we suppose the angular velocity of pin B about center C to be uniform and equal to unity, the crank radius CB will represent the velocity of its pin and, to the same scale, the distance CD , the velocity of the slide A at the same instant. Then the sides of triangle BCD will replace the sides of BPA as the representatives of the velocities. The length DB thus represents on our scale the aforesaid velocity of point A of the rod when turning about its point B .

We are now ready to determine the acceleration of point A . When the direction of an acceleration and the intensity of one of its two components and the direction of both are known, we can easily find the intensity of the acceleration itself by means of the triangle of accelerations, (which triangle is analogous to that of velocities or forces). In this case we know the direction of the acceleration of the point A to be along its path CAL . We may suppose it to be resolved into two components, one along the rod AB and the other at right angles thereto. We will determine the one along the rod AB . It was shown above that the motion of A was compounded of its own motion about B and of B 's motion. Now when the crank has a uniform angular velocity equal to unity its acceleration is represented in direction and intensity by BC . If we resolve BC into components along and at right angles to BA , the one along the rod will be BC' . The part of the acceleration of A which is due to its rotation about B as a center is also made up of two components, one along AB and the other at right angles to AB . We need to know only the former,

* This can also be regarded as an example of the resolution of a rotation about one axis P into an equivalent rotation about another parallel axis B plus a circular translation. See Weisbach-Herrmann's *Machinery of Transmission*, Vol. III, Sect. II, § 4 of the Introduction.

and this is $\frac{(\text{velocity})^2}{AB}$. But it was shown above that this velocity $= BD$, hence the component along AB is equal to $\frac{BD^2}{AB}$. This quotient can be found geometrically by describing a circle on the rod AB as a diameter and finding its intersections with an arc struck from B as a center and BD as a radius. The common chord IJ joining these intersections will cut from BA the distance $BF = \frac{BD^2}{AB}$ and will also cut from the horizontal through C , a distance CH exactly equal to the desired total acceleration of slide A . This can now be easily shown, for the component BF' just found has a direction from F' to B and the component BC' of B along the rod has evidently the direction B to C' . Therefore $BC' - F'B = F'C =$ that one of the two components of the total acceleration of A which acts along rod AB . Through C draw CF equal and parallel to $C'F'$ and to it at F erect a perpendicular FH ; this cuts from ACB , the direction of the total acceleration of A , a distance HC equal to the direction and intensity of the total acceleration of slide A .*

The point F' can also be found by drawing through D , the intersection of the rod AB with CY the perpendicular to the stroke, the parallel DE and where this meets the crank draw EF' parallel to CY ; this parallel cuts the rod or its prolongation in the desired point F' . The proof of this construction is:

$$BF' : BD = BE : BC = BD : BA, \text{ that is, } BF' = \frac{BD^2}{AB}.$$

* This construction is not only the simplest, but it is also applicable to all slider-crank mechanisms, whether the stroke of the slide A does or does not pass through the crank center C . In the latter as in the former case the intercept must be taken on a line CH through C and parallel to the slide stroke. It is also applicable to variable crank-pin velocity when scale is such that latter is represented by crank AB . We have then only to draw a parallel to stroke through the end of crank-pin acceleration (which in this case does not fall at center C of shaft). On this parallel the intercept included between the end of acceleration and chord IJ , Fig. 14, will be the exact acceleration of slide A .

But this construction of F' evidently fails when the crank is at either dead point.

The determination of the exact acceleration of A will enable us to construct its curve of acceleration NOQ , which is drawn on the stroke of A as a base. This suffices for most of the cases arising in practice. Greater accuracy is of course obtainable if we ascertain the acceleration of each point of the rod, multiply this by the elementary mass at that point and then find the resultant of all these elementary forces of inertia. This resultant can be resolved into two components, one acting at wrist pin A and the other at crank pin B . These can then be combined with the other forces acting at A and B and the exact tangential pressure on crank pin found. We will postpone this combination of forces for the present and confine ourselves to finding an easy method of constructing the acceleration of each point of the rod and its components parallel to, and perpendicular to, the rod.

We will first show that the end of the acceleration KM of any point K of the rod lies on the straight line joining the ends C and L of BC and AL , the accelerations of the two points B and A . To do this we must make use of the properties of the center of acceleration G .^{*} This center is that point of the moving rod which has no acceleration. The acceleration of any point K is directly proportional to the distance GK of this point

^{*} For method of finding the center of acceleration G when the directions of the acceleration of two points are given and also their ratio, see Weisbach-Herrmann's *Machinery of Transmission*, Vol. III., Section I, § 21. But for this, most common case, in which the crank has uniform rotation, we would suggest connecting crank-pin center B with point H (the intersection of line IJ with horizontal CB , Fig. 14) and on this connecting line laying off from B a distance equal to the length of connecting rod; then through the end of this distance draw a horizontal till it cuts the prolongation of the crank. The triangle thus obtained will be similar to CBH and when this triangle is swung around B till its base coincides with BA its vertex will fall at G and give exactly the position of the center of acceleration. The modifications necessary for cases of variable rotation of crank are evident; the triangles must be similar.

from the center of acceleration G , and for the instant in question the acceleration of each point makes the same angle with its own instantaneous radius of acceleration. For example, let us suppose the center of acceleration G known, and let BC , AL and KM represent respectively the accelerations of points B , A and K in direction and intensity.

Then will $BC:AL:KM = GB:GA:GK$
and angles $CBG = LAG = MKG$.

From these properties, it follows that the triangles GBC , GKM and GAL are all similar and hence their corresponding angles BGC , KGM and AGL are equal to each other. If we turn the whole system of points AKB about the center G through the angle $BGC = KGM = AGL$, the points B , K and A will fall on B' , K' and A' . Now if we can show that MC is parallel to $K'B'$ and ML to $K'A'$, we shall have proved that MC and ML are one and the same straight line because $K'B'$ and $K'A'$ constitute one straight line. From the similarity of the triangles GBC , GKM and GAL , we have

$$\begin{aligned} GC:GM &= GB:GK = GB':GK' \\ GM:GL &= GK:GA = GK':GA' \end{aligned}$$

and therefore MC and ML are respectively parallel to $K'B'$ and $K'A'$ and we have proved that the ends of all the accelerations of the points of one straight line lie on one and the same straight line.

This is true of any line of the system and hence of any combination of lines belonging to or constituting the system. Moreover it is evident that any line in the plane of rotation will be reduced in the ratio of GC to GB so that

$$\frac{CL}{BA} = \frac{GC}{GB}$$

The ends of the accelerations of all points in the plane figure constituting the rod will form a reduced *image* of the shape of the rod which will be exactly similar (in the plane of motion) to the original rod. This similarity extends to every detail.

A corollary from this proposition is that the ends of the accelerations of the various points divide the distance CL in the same ratio as do their corresponding points the rod or distance $BA = B'A'$. For instance, we have

$$CM : ML = B'K' : K'A' = BK : KA.$$

This corollary enables us to determine the acceleration of any point of the rod AB (without the help of the center of acceleration G) when we know the direction and intensity of the accelerations of any two points on that line. For example, if K is half way between A and B , the end M of its acceleration KM will be half way between the ends C and L of the known accelerations BC and AL .

If the total accelerations of each of the points of the rod be resolved into two components, one parallel and the other perpendicular to the rod, and these components be laid off as ordinates, (taking the rod as an axis of abscissas), one set of components on one side of the rod and the other set on the opposite side of the rod, then will the components of each set terminate on the same straight line. That is, the components of the set which are at right angles to AB will terminate on the straight line $A''B''$, and the components of the set which coincide with AB , will (when revolved through 90°) terminate on the straight line RST .

This can be proved as follows: Drop from C , M and L the respective perpendiculars CC' , MM' and LL' . Because they are a series of parallels they will divide the distance CL' in the same ratio as CL , and since CL has been shown to be divided by the acceleration in the same ratio as BA , it follows that CL' is divided in the same ratio as BA . But the ordinates CC' , MM' , LL' , etc., erected on CL' , by construction, terminate in a straight line, namely CML . As AF' is equal to $L'C$ we may suppose these self-same ordinates to be laid off from the former and then they will all terminate on the straight line $A''F$. Now if we suppose the base AF' of this set of ordinates to be stretched till it is equal to AB in length, the ordinates remaining equidistant and chang-

ing only their position and not their magnitude, then will they all terminate on the straight line $A''B''$. For before stretching, the ratio $\frac{dy}{dx} = a$, of the difference of any two adjacent ordinates to the difference of their abscissas, was equal to a constant a , which is the characteristic of a straight line. After stretching, the ratio of these differences is $\frac{dy}{dx^2} = a' = \text{a constant also, because } \frac{dx^2}{dx} = \frac{AB}{AF} = \text{a constant for the points of this rod and this position of the mechanism.}$

If we resolve the accelerations BC, KM, AL , etc., through 90° so that they occupy positions BC'', KM'', AL'' , etc., these new positions will also make a constant angle with their instantaneous radii of acceleration, and we may show as before that the points $L''M''C''$ lie on a straight line. If from their ends C'', M'' and L'' we drop on AB the perpendiculars $C''C', M''M', L''L'$, etc., and erect these as ordinates at the corresponding points of the base AB we can show as before that these all terminate on one straight line RST . But as these perpendiculars $C''C'$, etc., are respectively equal to the second set of components BC , etc., the second part of our proposition has been established. The forces of inertia due to this second set of components do not however act transversely on the rod, but longitudinally. Their sum or resultant combined with the resultant of all the bending forces that are due to inertia will give a total resultant, due to inertia of rod, that can be resolved into two forces acting at pins A and B .

Exactly how this resultant is found will be discussed later on; at present the shaded area and area $ABTR$ furnish the components for constructing the acceleration of any point belonging to the center line of rod. For this purpose however area $ABTR$ would suffice, for if its component be laid off on the rod, say $KM = KS$, a perpendicular erected at M will cut the line CA in J giving at once KM as the acceleration of the point K .

III A.

ACCELERATION DIAGRAMS.

Obtain by means of the preceding diagrams or tables, or both the accelerating and retarding forces corresponding: to an assumed value of $\frac{F_o}{A}$, to a given ratio of connecting rod to crank, and to certain crank angles. Then find the corresponding piston positions as in the preceding plate. At these positions erect ordinates equal to the accelerating or retarding forces the former being laid off below, and the latter above, the line of piston positions. It is evident from the figure that the acceleration curve for one stroke can be obtained from that of the other by revolving the latter 180° about the central line.

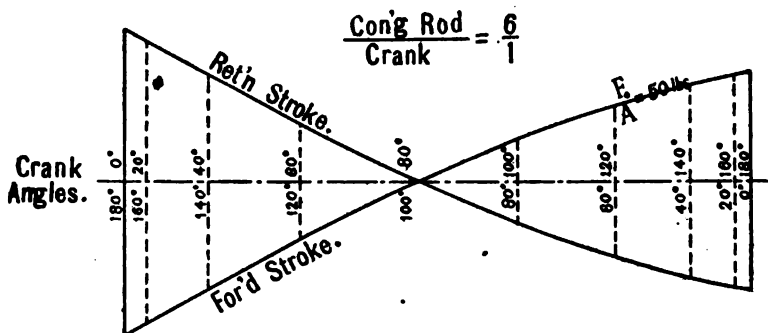


Fig. 17.

Construct on the admission line of each of the six diagrams of effective steam pressure on piston, two acceleration diagrams (in the manner shown in Fig. 17) which correspond respectively to $\frac{F_o}{A} = \frac{p_1 - 16}{2} = \text{say, } 40 \text{ lbs.}$ and $\frac{F_o}{A} = 2p_2 = \text{say, } 50 \text{ lbs.}$; p_2 is the terminal pressure at the end of the expansion curve. Assume also $\frac{F_o}{A} = 30$.

III B.

DETERMINATION OF PRESSURE ON CROSS-HEAD PIN, WHEN
WEIGHT OF ROD AND FRICTION ARE NEGLECTED, BUT
INERTIA TAKEN INTO ACCOUNT BY ASSUMING WHOLE
MASS OF ROD AND RECIPROCATING PARTS
CONCENTRATED ON PISTON.

To obtain this diagram we must combine the diagram of effective steam pressures on piston with the diagram of accelerating and retarding pressures. Moreover the combination must be so effected that the accelerating pressures will be subtracted from the effective steam pressures, and the retarding forces added. This can be done in two ways: by subtracting and adding the ordinates of the acceleration diagram, Fig. 17, to the upper lines *ABC* and *EFG* of Figs. 8 and 9, representing the pressures on driving side of piston, and getting as a result the horizontal pressure diagrams *VTSPH* and *G'POLA'DC'*, Figs. 18 and 19, or by subtracting and adding the ordinates of Fig. 17 to the lower lines *H'E* and *A'D'C'*, Figs. 8 and 9, representing the counter pressures, and getting as a result the horizontal pressure diagrams *A'B'C'MN* and *E'F'G'HKE'*, Figs. 20 and 21. A comparison of the two sets of diagrams will show that the corresponding ordinates included by the shaded portions are equal.

The shaded area in each case is bounded on the one side by the resultant line (due to the combination of expansion line or line of counter-pressure with acceleration curve) and on the other side by that line of Fig. 8 or 9 which did not help to form the resultant line.

It should also be noticed that where the resultant line crosses the unused line (given by Fig. 8 or 9) the horizontal pressures on cross-head pin become equal to zero, and after passing this point, see Fig. 21, the direction of the horizontal pressures on cross-head pin is reversed before the piston reaches the end of its stroke. In Fig. 20 the steam and retarding pressures are so related as to com-

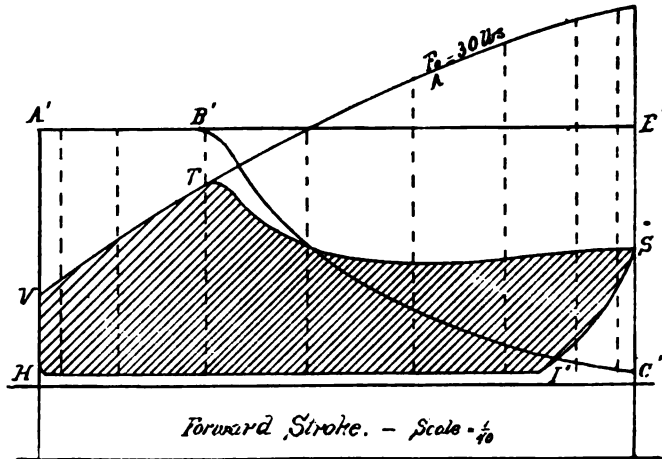


Fig. 18.

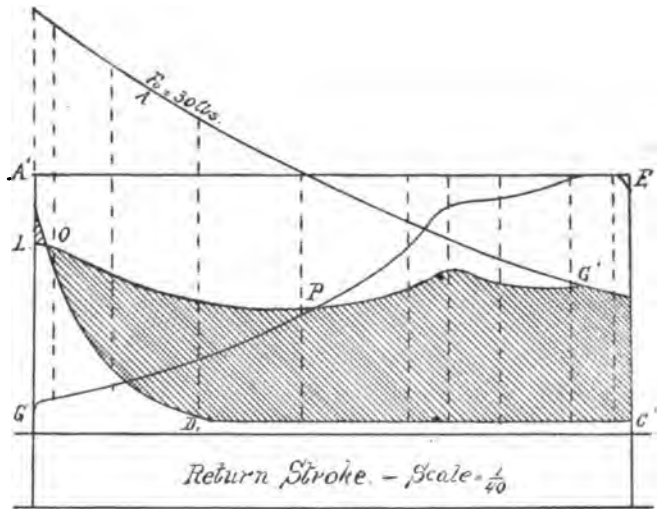


Fig. 19.

pletely unload the crank-pin at the end of the stroke. This is not so favorable as it seems, for at the beginning of the return stroke there is suddenly applied a great pressure in the opposite direction. A little more compression $I'S$, Fig. 18, would have changed this. It is probable that the arrangement shown in

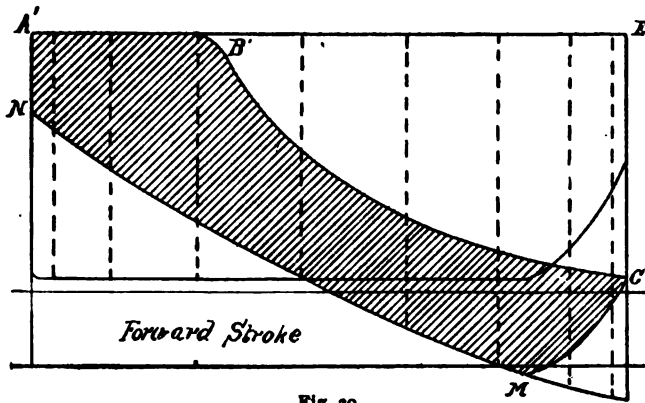


Fig. 20.

Fig. 21, is less likely to cause pounding than that in Fig. 18, and this in spite of the fact that in the case of Fig. 21 the reversal of pressure near end of stroke occurs while piston is in motion. A

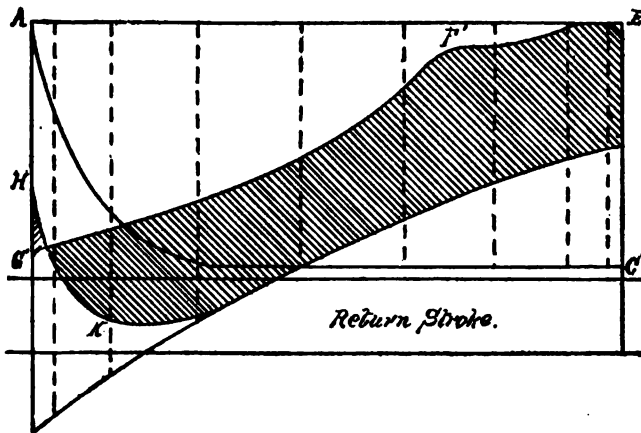


Fig. 21.

less lead in the case of Fig. 19 would be advantageous in cutting down the large initial pressure.

Another construction, used by Grashof, is well worth noting. It is to lay off these resultant pressures at cross-head pin on the rectified crank-pin circle. The straight base will then represent time, and the rate at which the pressure changes can then be

- easily seen. This will be particularly useful in judging of the rapidity with which the reversals of pressure occur.

We shall hereafter only make use of horizontal pressure or wrist-pin diagrams similar to those shown in Figs. 18 and 19.

III c.

EFFECT OF VALVE SETTING ON THE HORIZONTAL PRESSURE ON CROSS-HEAD PIN.

We give two indicator diagrams, Figs. 22 and 23, from opposite ends of the same cylinder to show the influence respectively of a late

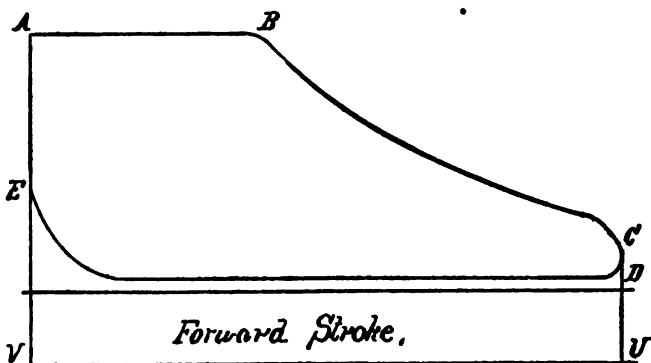


Fig. 22.

admission and too late an exhaust. By combining the diagram $ABCUV$, Fig. 22, with diagrams of counter-pressure HJK , Fig. 23, we get the effective steam pressure diagram $ABCKJIH$, Fig. 24.

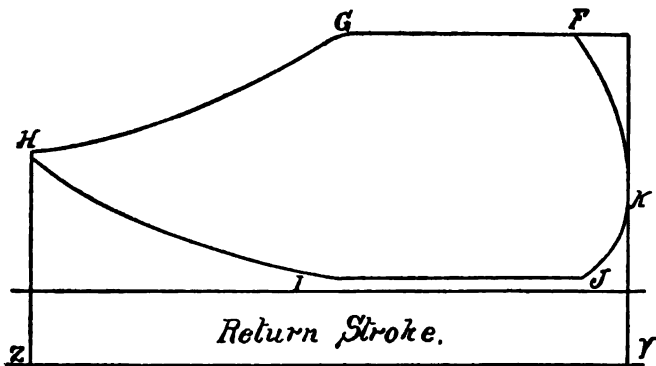


Fig. 23.

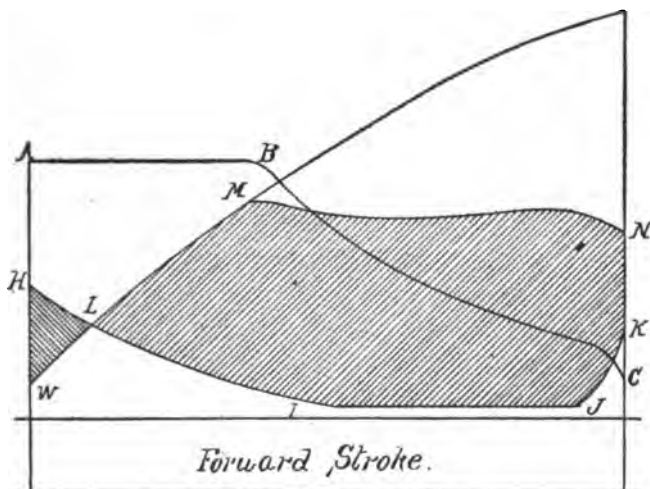


Fig. 24.

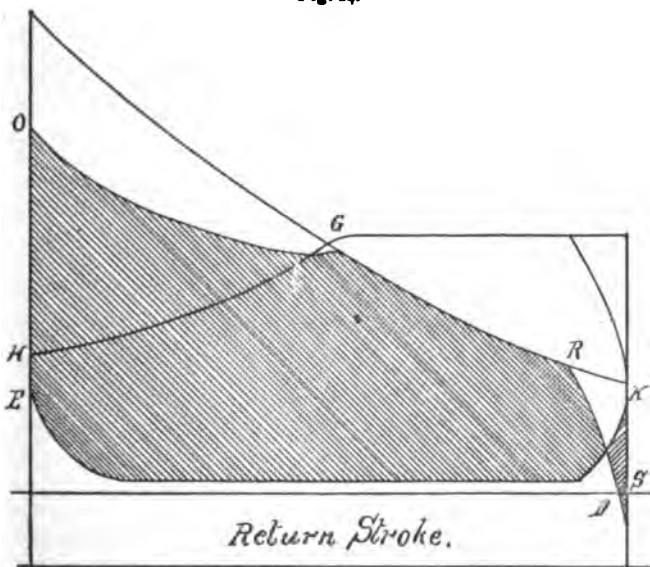


Fig. 25.

In like manner by combining the counter-pressure diagram $EDUV$, Fig. 22, with $HGFKYZ$, Fig. 23, we get the effective pressure diagram $KDEHG$, Fig. 25. By combining these real

pressure diagrams with the acceleration diagrams we get the horizontal pressures on cross-head pin, shown in Figs. 24 and 25 by shaded areas. Fig. 24 shows that, owing to the late exhaust *HI*, Fig. 23, there is a drag instead of a push on the crank during the first portion of the stroke, and a reversal of pressure on the crank while the piston is in rapid motion at *L*. This reversal will cause a shock or knock if there is any play in the joints of the mechanism. In like manner Fig. 25 shows a drag instead of a push on crank at beginning of stroke and a reversal of pressure at *D*, but here the piston motion is considerably slower than at *L*, so there will probably be no shock. The final pressure, *KS* Fig. 25, on piston at beginning of return stroke is about equal to that, *KN* Fig. 24, at end of forward stroke, and is a desirable result. It would be better however to accomplish the same result by more compression and earlier admission. Fig. 25 also shows the very unequal distribution of the horizontal pressures which attends a large cut-off when unaccompanied by suitable compression. By having the proper amount of compression at *S*, Fig. 18, not only may the reciprocating parts be brought to rest without shock, but all load can be taken off the crank-pin while it is passing the dead center. This is not always done on account of the decided advantages gained by compressing nearly up to the boiler pressure, some of which advantages are the filling and heating of the hurtful space by compressed steam instead of the live, fresh, steam and the avoidance of such admission lines as *KF*, Fig. 23. Moreover, the reversal of pressure which accompanies considerable compression is not necessarily an evil, provided it takes place when the piston is moving slowly, *i. e.* near the end of stroke where reversal naturally takes place. The ideal case is to have the piston unloaded while moving slowly (near end of stroke) and then let the driving pressure on opposite side increase gradually till after the reverse stroke has begun. Considerable compression and even a slight negative lead may be used to effect this desirable result, desirable, because of its freedom from pounding.

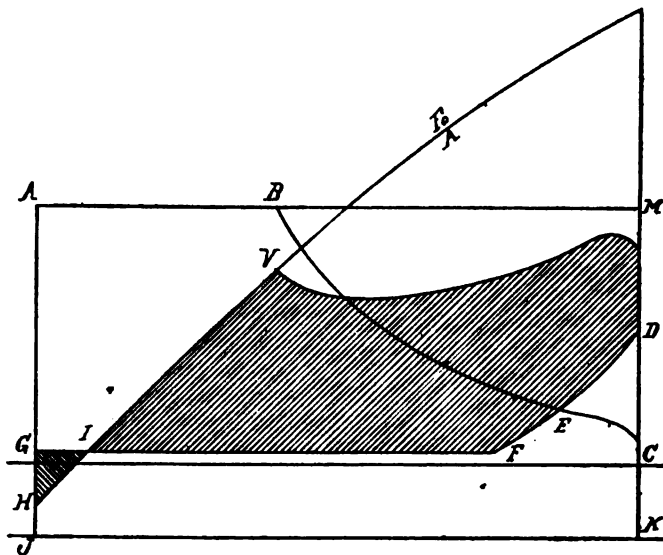


Fig. 26.

In Fig. 26 the ordinate AH at the beginning of the acceleration curve $\frac{F_o}{A}$ is greater than the corresponding effective steam pressure AG ; the result is that instead of there being a push upon the crank in the direction of its motion at the beginning of the stroke, there is a drag upon the crank, this condition continuing until the piston reaches the position I , where a reversal of pressure takes place causing shock and vibrations injurious to the durability of the machine, because at that end of the stroke there is considerable piston speed at point I .

In Fig. 27 the condition of affairs is still worse, for two reversals of pressure take place at I' and I'' , causing two shocks in rapid succession. In this last case, although the effective pressure $A'G'$ is greater than the initial pressure of acceleration $A'H'$, the ordinates $\left(= \frac{F}{A} = \frac{F_o}{A}(\cos \omega + \frac{R}{L} \cos 2\omega) \right)$ of the acceleration

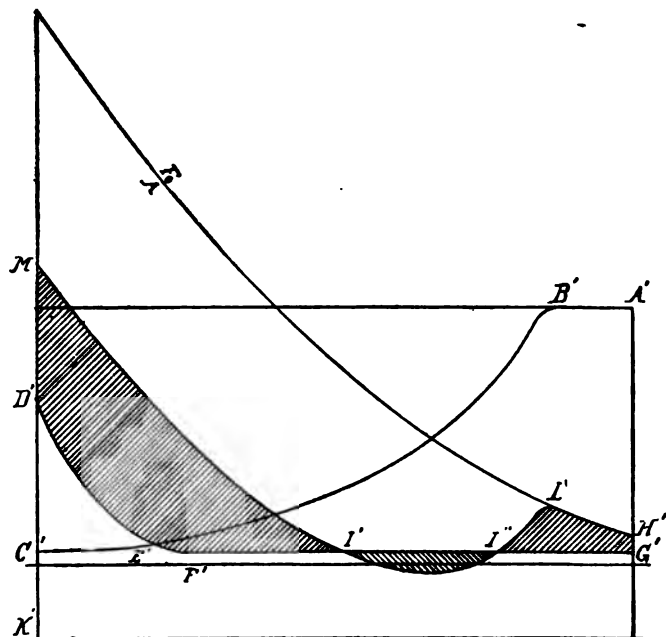


Fig. 27.

curve $\frac{F_o}{A}$ are still too large for the short cut-off $\frac{A'B'}{G'C'}$ and the small initial pressure $A'G'$. It is evident from the figures that if the effective pressure diagrams $ABCDEFGA$ and $A'B'C'D'E'F'G'A'$ of Figs. 26 and 27 are to remain unchanged, that is, if the mean effective pressure pin is to remain unchanged, the only remedy for the evils exhibited in Figs. 26 and 27 is to diminish the value of $\frac{F_o}{A}$.

When the engine varies between wide limits of cut-off, as in the present case, the value of $\frac{F_o}{A}$ is chosen so as to favor as much as possible the particular horse power at which the engine ordinarily runs; in such a case it is desirable to know what is the minimum cut-off at which reversal of pressure will be avoided. This may be ascertained by the cut and try method, but it may

also be calculated with sufficient accuracy if we make the following suppositions, namely, that the curve of expansion is an equilateral hyperbola, that the best value for $\frac{F_o}{A}$ is

$$\frac{F_o}{A} = p_1 - \frac{16}{2} *$$

and that the crank angle is about 60° when the horizontal pressure curve EFG , (Fig. 28), is tangent to the back pressure line

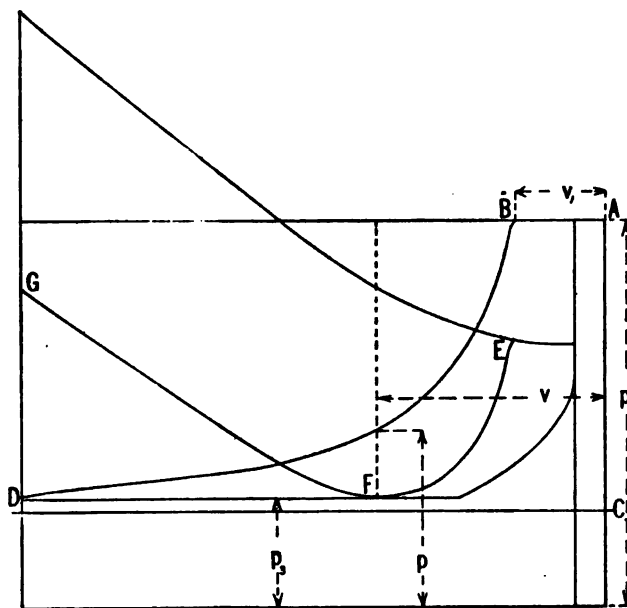


Fig. 28.

DF. As the reversals in question are more apt to occur during the return stroke we must use a formula

$$p - p_3 = \frac{F_o}{A} \left(\cos \omega - \frac{R}{L} \cos 2\omega \right) = \frac{v_1}{v} p_1 - p_3 \quad (40)$$

corresponding to that stroke and introduce the assumptions just made.

* Radinger holds that $\frac{F_o}{A} = 2p_2$ = twice the terminal pressure at end of expansion will give the steadiest running of engine.

By deriving a value for v from the piston travel s for return stroke and allowing for clearance, we find that under the average conditions of running of high speed engines, the minimum, real, cut-off should be greater than 0.10 to avoid reversals of pressure when piston is running swiftly. The approximate formula for this cut-off is:

$$\frac{AB}{DC} > \frac{1}{8p_1} \frac{F_o}{A} \left(\frac{L}{R} + 1 \right) + \frac{p_3}{4p_1} = \left(\frac{1}{16} - \frac{1}{p_1} \right) \left(\frac{L}{R} + 1 \right) + \frac{4}{p_1} \quad (41)$$

To find the number of horse powers corresponding to this minimum, real, cut off we look in Table III for value of $\frac{p''_{m1}}{p_1}$ and then transforming, Eq. 14, we get

$$\begin{aligned} H_i &= \left[1 - (1 + i) \left(\frac{p''_m}{p_1} - \frac{p''_{m1}}{p_1} \right) \frac{p_1}{p_m} \right] H \\ &= \left[1 - (1 + i) \left(\frac{p''_m}{p_m} - \frac{p''_{m1}}{p_m} \right) \right] H \end{aligned}$$

All the terms of the second member being known we can calculate readily the minimum horse power at which the engine can be safely run at the given speed and pressure.

Find minimum cut off and horse power for present engine. Construct on each of the six diagrams of effective steam pressures on piston two diagrams of resultant pressures on cross-head pin corresponding to $\frac{F_o}{A} = p_1 - \frac{16}{2}$ and $\frac{F_o}{A} = 2p_1$, i.e. $\frac{F_o}{A} = 50$ and 40 lbs.

III D.

CONVERSION OF HORIZONTAL PRESSURES ON CRANK PIN INTO TANGENTIAL PRESSURES.

The horizontal component of the pressure of the rod against the crank pin is evidently equal to the horizontal pressure of the cross-head pin. In this problem we are to find the rotative effect of a horizontal pressure on crank pin.

Let P^1 represent horizontal pressure on cross-head pin.

T represent force along connecting rod.

t represent tangential pressure on crank pin.

then will
$$T = \frac{P^1}{\cos a} \quad (42)$$

and
$$t = T \sin (\omega + a) = P^1 \frac{\sin (\omega + a)}{\cos a} \quad (43)$$

$$\frac{t}{P^1} = \frac{\sin (\omega + a)}{\cos a} \quad (44)$$

If we substitute in this equation $\sin a = \frac{R}{L} \sin \omega$ and $\cos a = \sqrt{1 - \frac{R^2}{L^2} \sin^2 \omega}$ (see Eqs. 21 and 22) we shall have the means of calculating the ratios $\frac{t}{P^1}$ from the crank angles ω . Table VIII on page 60 contains these ratios.

If we wish to determine the tangential pressures from given horizontal pressures by graphical means we can proceed as follows, Fig. 29.

In triangle AKC , Fig. 29, we have

$$\frac{KC}{AC} = \frac{KC}{R} = \frac{\sin (\omega + a)}{\cos a} \quad (45)$$

or

$$\frac{t}{P^1} = \frac{KC}{R} = \frac{w}{v} \quad (46)$$

hence if on any scale, say 20 lbs. to the inch, we lay off $DE = P^1$, join K with D and prolonging KD till it intersects at F the perpendicular to DE erected at E , we will have in the two similar triangles, DKC and DFE , the proportion $FE : P^1 :: KC : R$, i. e. $t = FE$.

TABLE VIII.

ROTATIVE EFFECT OF A UNIT OF HORIZONTAL PRESSURE ON CRANK.

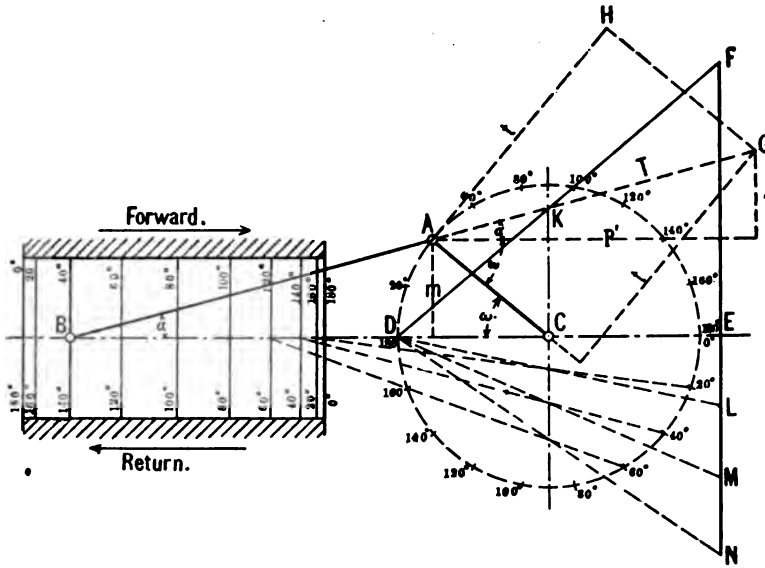
To find actual tangential pressure (or rotative effect) on crank, multiply tabular quantity by resultant horizontal pressure on crank or cross-head pin.

Forward stroke is towards, and return stroke away from, crank shaft.

To find wrist-pin velocity multiply tabular values by crank-pin velocity.

Crank Angles.		Connecting Rod ÷ Crank ==					
Forwa'd.	Return.	4.0	4.5	5.0	5.5	6.0	∞
5	175	.1089	.1064	.1045	.1030	.1016	.0832
10	170	.2164	.2117	.2079	.2047	.2022	.1737
15	165	.3215	.3145	.3089	.3054	.3005	.2588
20	160	.4227	.4136	.4065	.4019	.3957	.3420
25	155	.5189	.5081	.4995	.4925	.4866	.4226
30	150	.6091	.5968	.5870	.5791	.5724	.5000
35	145	.6923	.6788	.6682	.6596	.6523	.5736
40	140	.7675	.7533	.7421	.7329	.7253	.6428
45	135	.8341	.8195	.8081	.7988	.7910	.7071
50	130	.8914	.8771	.8657	.8564	.8488	.7660
55	125	.9392	.9253	.9144	.9055	.8982	.8192
60	120	.9769	.9641	.9540	.9458	.9390	.8660
65	115	1.0046	.9932	.9842	.9769	.9709	.9063
70	110	1.0224	1.0127	1.0052	.9990	.9939	.9397
75	105	1.0304	1.0228	1.0169	1.0121	1.0082	.9659
80	100	1.0289	1.0237	1.0199	1.0164	1.0137	.9848
85	95	1.0186	1.0160	1.0139	1.0127	1.0109	.9962
90	90	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
95	85	.9738	.9764	.9785	.9797	.9816	.9962
100	80	.9407	.9460	.9500	.9532	.9559	.9848
105	75	.9016	.9091	.9150	.9198	.9237	.9659
110	70	.8571	.8667	.8743	.8804	.8855	.9397
115	65	.8080	.8194	.8285	.8357	.8418	.9063
120	60	.7552	.7680	.7781	.7863	.7931	.8660
125	55	.6992	.7130	.7239	.7328	.7401	.8192
130	50	.6407	.6550	.6664	.6756	.6833	.7660
135	45	.5801	.5958	.6061	.6155	.6232	.7071
140	40	.5181	.5323	.5435	.5527	.5603	.6428
145	35	.4549	.4683	.4790	.4876	.4949	.5736
150	30	.3909	.4032	.4130	.4209	.4276	.5000
155	25	.3264	.3371	.3458	.3528	.3586	.4226
160	20	.2614	.2704	.2776	.2821	.2884	.3420
165	15	.1962	.2032	.2088	.2123	.2171	.2588
170	10	.1309	.1356	.1394	.1426	.1451	.1737
175	5	.0655	.0679	.0698	.0714	.0727	.0872

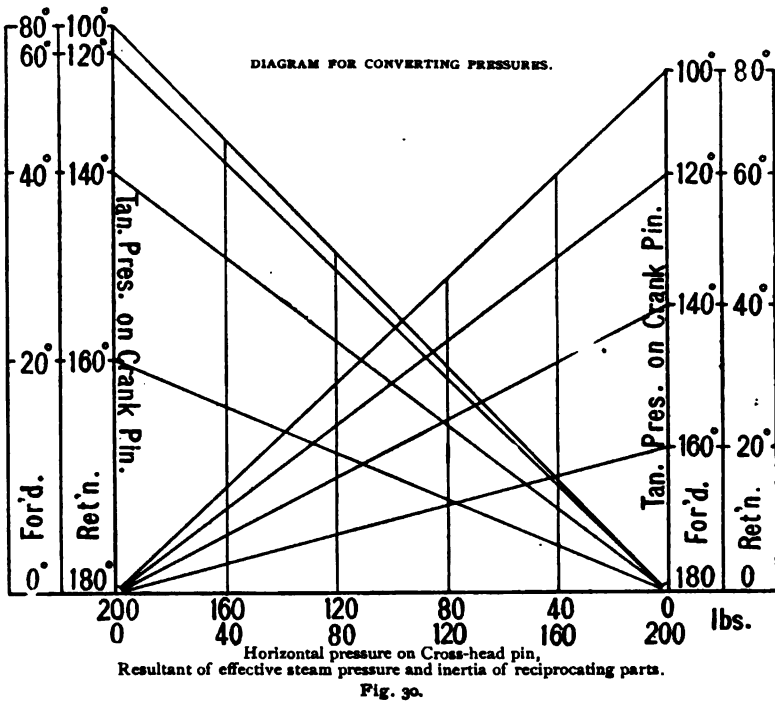
The value of t will be on the same scale as P^1 . If we determine in like manner, for the same pressure $P' = DE$, the tangential pressures corresponding to the various crank positions of Fig. 29 and lay them off on the extreme left or right hand ordinates of



the section paper employed, then drawing diagonals to the assumed zero of cross-head pin pressures we will get a diagram similar to Fig. 30 from which we can determine the tangential pressures for any value of P^1 . In Fig. 29 the ratio of length of connecting rod to length of crank was taken $2\frac{1}{2}$. In the diagram to be drawn on section paper assume this ratio to be $= 6$, also $DE = P^1 = 200$ lbs.

Lay off, in Fig. 30, the tangential pressures on a scale of 20 lbs. to the inch, and the horizontal or cross-head pressures on a scale of 20 lbs. to the inch. The tangential pressures can also be obtained in the ordinary manner by resolving the force acting along the connecting rod into two components respectively tan-

gent and normal to the path of the crank-pin. But this would involve a good deal more labor than the present method. Figs. 29 and 30 are quite accurately drawn and the method of obtaining 30 from 29 can therefore be easily followed. Thus *EL*, *EM* and *EN* are the tangential pressures corresponding respectively to the angular positions 20° , 40° and 60° of crank for *return*



stroke and to 160° , 140° and 120° for the *forward stroke*. The quickest method of obtaining a diagram like Fig. 30 would be to calculate the tangential pressures from Table VIII, assuming a constant horizontal pressure of 200 lbs. on cross-head pin. It is sufficiently evident from Fig. 30 how the tangential pressures for other and smaller resultant pressures on cross-head pin can be obtained.

III E.

CONSTRUCTION OF DIAGRAM OF TANGENTIAL PRESSURES ON
CRANK-PIN.

We are now ready to construct the tangential pressure diagrams from the diagrams of resultant pressures on cross-head pin corresponding to various values of $\frac{F_o}{A}$. We proceed as in the construction of Fig. 13. By means of the diagram, Fig. 30, we first transform the horizontal pressures into tangential pressures, and then lay off these tangential pressures as prolongations of the radii of the crank-pin circle (or of any other convenient circle), the latter forming the base of the radial ordinates. A curve is next drawn through the extremities of these ordinates and then a circle concentric with—and outside of—the crank-pin circle, is drawn, the difference between the two circumferences is equal to the mean tangential pressure, which as before is equal to $p_t = 0.6366 p_m$, Eq. 17. The deviations of the curve (drawn through the extremities of the ordinates representing the tangential pressures) from the circle of tangential pressures will show the irregularity of the driving power. The more closely the tangential pressure curve approximates to the mean-tangential-pressure-circle, the steadier will the engine run. Figs. 31 and 32 show two different methods of representing the tangential pressures. In the former figure the tangential pressures are laid off from the crank-pin circle itself; in the latter from the rectified semi-circle. Fig. 31 has the advantage of showing the phases of resistance $CMNO$ and AQS and the curve of tangential pressures in a continuous manner, but it is subject to the following very slight disadvantage, viz: that the area enclosed by $AQRM CBA$ does not represent with great accuracy the work done in a semi-revolution, the inaccuracy being due to the divergence of the radial ordinates. The objection does not apply to Fig. 32, when the area $AVECBA$ is exactly equal to the

area $AFDCA$ = work done in one semi-revolution; moreover by adding the shaded portions $CD'E'$ ($= CDE$) and AFG' as shown in Fig. 32, the objection that this mode of representation does not represent the phases of resistance in a continuous man-

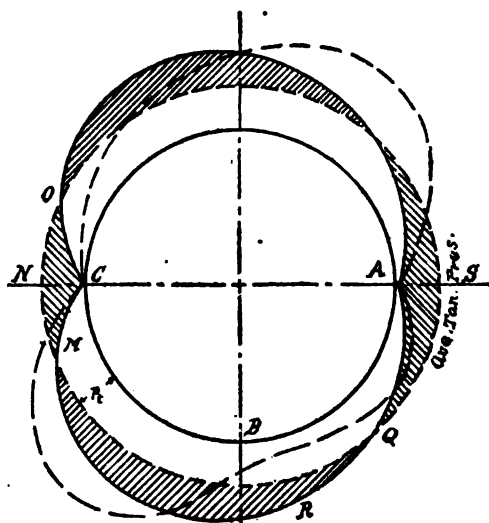


Fig. 31.

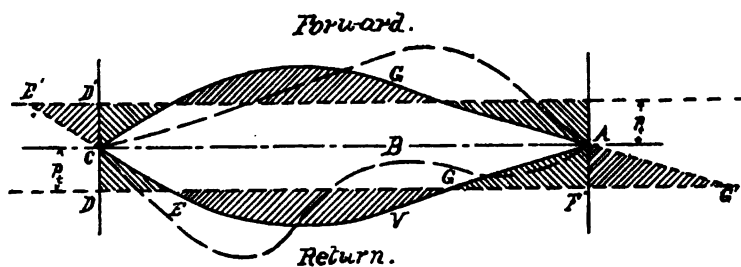


Fig. 32.

ner disappears. If the work has been done correctly it will be found that in Fig. 31 the area of the shaded portion QRM = area CMN + area QAS , nearly, and that in Fig. 32 the phase of excess $GVE = AGF' + DEC$ exactly.

Inspection of either of these figures shows that the tangential pressure line (shown by full line) represents steadier running than

the tangential pressures corresponding to greater piston speed or heavier reciprocating parts (shown by broken line).

Draw for the minimum, rated and maximum power of the proposed engine, separately two tangential pressure diagrams, one like Fig. 31 and the other like Fig. 32. On each diagram draw two tangential pressure curves corresponding to the resultant, or horizontal, pressure on cross-head pin which were drawn for the assumed values: $\frac{F_o}{A} = \frac{p_1 - 16}{2} = \text{about } 40$ and $\frac{F_o}{A} = 2p_2 = \text{about } 50$ lbs.

When these have all been drawn, inspect the result and see if a different value than those chosen for $\frac{F_o}{A}$ would not give a more uniform tangential pressure—i.e. steadier motion. If so,

take the new value of $\frac{F_o}{A}$ and draw corresponding acceleration curves for both the forward and the return stroke, and superimpose these upon the diagrams of effective steam pressures already drawn and combine them as before to form diagrams of resultant pressures on cross-head pin. Then convert these last diagrams into tangential pressure diagrams and again inspect the latter diagrams to see if any improvement can be effected. This process is to be continued till that value for $\frac{F_o}{A}$ has been found which

gives the most uniform tangential pressure for the normal, rated, horse power of the proposed engine. This most favorable value

of $\frac{F_o}{A}$ for the rated power should however be examined with reference to the minimum power required, to see whether it then causes reversals of pressure. When this is the case some smaller

value of $\frac{F_o}{A}$ less favorable to the normal or rated power should be chosen for the engine. In this connection we may quote the following from *Rigg's Steam Engine*.

CHOICE OF SPEED AND LOAD. •

“There are, then, three elements which can be adjusted to each other, namely, steam pressure, including the rate of expansion, weight of reciprocating parts, and number of revolutions per minute, or piston velocity; any of these elements can be altered at the option of the designer of the engine, and the problem to be solved is not to transfer the strains from one end of the stroke to the other, not to work steadily with the highest expansion, not to produce a regular uniform horizontal pressure on the crank. But the end and object of all these calculations and changes in the old acknowledged rules for the construction of engines is, by working with that amount of expansion which practical experience proves economical, to obtain as nearly as possible a uniform tangential pressure on the crank. When this is done an engine will drive its load steadily and well, and the influence of the reciprocating parts has a most direct bearing on this most important subject.

In applying the foregoing reasoning it is necessary to exercise a judicious choice in balancing the evils which arise from a too low or too high speed, and so to decide a rate at which an engine can be run to the best advantage. When the load upon the engine is regular it is comparatively easy to do this, but in a great majority of cases the load is often varying, and the speed is bound to remain constant. Thus, it is necessary to select a load on the engine which shall meet most requirements, and then the indicator diagram either actually taken or assumed will form the principal datum required.”

To these considerations might be added another, namely, that to avoid pounding or shocks at crank- or wrist-pin, the driving pressures should change as gradually as possible from one side to the other of piston, the change taking place as near the dead point as possible. By using considerable compression or even a slightly negative lead this gradual change may be accomplished for both ends of the stroke.

IV.

DETERMINATION OF DIAMETER OF CYLINDER D , LENGTH OF STROKE S , REVOLUTIONS PER MINUTE N OR PISTON SPEED PER MINUTE w^s AND WEIGHT OF RECIPROCATING PARTS W BY MEANS OF THE TABLES OR FORMULAS.

$$\frac{F_o}{A} = 0.0000142 \frac{W}{A} S N^2 = .000085 \frac{W}{A} N w^s$$

$$\frac{F_o}{A} = 3.63 \frac{H}{p_m} \frac{W}{A} \frac{N}{D^2} = 2.18 \frac{w^s}{S D^2} \frac{W}{A} \frac{H}{p_m} \quad (47)$$

$$\frac{H}{p_m} = 0.0000238 w^s D^2 \quad (48)$$

$$\frac{H}{p_m} = 0.00000396 N S D^2 \quad (49)$$

$$\frac{\frac{F_o}{A}}{\frac{H}{p_m}} = 3.60 \frac{W}{A} \frac{N}{D^2} = 21.6 \frac{W}{A} \frac{w^s}{S D^2} \quad (50)$$

H = number of indicated H. P.

S = length of stroke in inches.

$\frac{W}{A}$ = weight of reciprocating parts per $\square''$ of piston, weight of rod included.

D = diameter of cylinder in inches.

w^s = average speed of piston in feet per minute.

N = revolutions per minute.

p_m = mean effective pressure per $\square''$ of piston.

$\frac{F_o}{A}$ = average force accelerating reciprocating parts when crank is on dead centers.

TABLE IX. *

VALUE OF $\frac{H}{p_m}$, OR INDICATED H. P. FOR EACH POUND OF MEAN
EFFECTIVE PRESSURE PER $\square''$ OF PISTON AREA.

Quantities in table were obtained by neglecting area of piston rod and assuming that the same mean effective pressure (p_m) existed on both sides of the piston area; consequently the tabular quantities are about $1\frac{1}{2}\%$ and $\frac{1}{2}\%$ too large for small and large cylinders respectively. To correct for piston rod multiply tabular quantities by $[1 - \frac{1}{2}(\frac{d^2}{D})^2]$ where D = diameter of cylinder and d = diameter of piston rod.

Diameter of Cylinder.	Average speed of piston in feet per minute.										
	700	750	800	850	900	950	1000	1050	1100	1150	1200
6	.600	.643	.685	.728	.771	.814	.857	.900	.942	.985	1.028
6½	.704	.754	.804	.855	.905	.955	1.006	1.056	1.106	1.156	1.207
7	.816	.875	.933	.991	1.050	1.108	1.166	1.225	1.283	1.341	1.399
7½	.937	1.004	1.071	1.138	1.205	1.272	1.339	1.406	1.473	1.540	1.607
8	1.066	1.142	1.219	1.295	1.371	1.447	1.523	1.599	1.676	1.752	1.828
8½	1.204	1.290	1.376	1.462	1.548	1.634	1.719	1.806	1.892	1.977	2.063
9	1.349	1.446	1.542	1.639	1.735	1.831	1.928	2.021	2.121	2.217	2.313
9½	1.504	1.611	1.718	1.826	1.933	2.041	2.148	2.255	2.363	2.470	2.578
10	1.666	1.785	1.904	2.023	2.142	2.261	2.380	2.499	2.618	2.737	2.856
10½	1.874	2.008	2.142	2.276	2.420	2.544	2.678	2.811	2.945	3.079	3.213
11	2.016	2.160	2.304	2.448	2.592	2.736	2.880	3.024	3.168	3.312	3.456
11½	2.203	2.361	2.518	2.675	2.833	2.990	3.148	3.305	3.462	3.620	3.777
12	2.399	2.570	2.742	2.913	3.084	3.256	3.427	3.599	3.770	3.941	4.113
13	2.816	3.017	3.218	3.419	3.620	3.821	4.022	4.223	4.424	4.626	4.827
14	3.265	3.500	3.732	3.965	4.198	4.432	4.665	4.896	5.231	5.365	5.597
15	3.749	4.016	4.284	4.552	4.820	5.087	5.355	5.623	5.891	6.158	6.426
16	4.265	4.570	4.774	5.179	5.484	5.788	6.093	6.397	6.696	7.007	7.311
17	4.815	5.159	5.503	5.846	6.190	6.534	6.878	7.222	7.566	7.910	8.254
18	5.397	5.783	6.169	6.555	6.940	7.326	7.711	8.097	8.482	8.868	9.253
20	6.664	7.140	7.616	8.092	8.568	9.044	9.520	9.996	10.472	10.948	11.424
22	8.063	8.639	9.215	9.791	10.367	10.943	11.519	12.095	12.577	13.147	13.723
24	9.596	10.282	10.967	11.652	12.338	13.023	13.709	14.394	15.080	15.765	16.451
26	11.162	12.067	12.871	13.675	14.480	15.284	16.089	16.893	17.698	18.502	19.307
28	13.061	13.984	14.927	15.860	16.793	17.726	18.659	19.592	20.525	21.458	22.391
30	14.994	16.067	17.136	18.207	19.278	20.349	21.421	22.491	23.562	24.633	25.704
32	17.060	18.278	19.497	20.716	21.934	23.153	24.371	25.590	26.798	28.027	29.245
34	19.259	20.635	22.010	23.386	24.762	26.137	27.513	28.888	30.264	31.640	33.035
36	21.591	23.134	24.676	26.218	27.760	29.303	30.845	32.386	33.929	35.472	37.014
38	24.057	25.775	27.494	29.212	30.930	32.648	34.367	36.086	37.804	39.522	41.239
40	26.656	28.500	30.464	32.368	34.272	36.176	38.080	39.984	41.888	43.792	45.696

* Table IX continued on page 69.

VALUE OF $\frac{H}{p_m}$, OR INDICATED H. P. FOR EACH POUND OF MEAN EFFECTIVE PRESSURE PER $\square''$ OF PISTON AREA.

This part taken from Riggs' Steam Engine.

Diam. of Cylinder	Average speed of piston in feet per minute.									
	240	300	350	400	450	500	550	600	650	750
4	.091	.114	.133	.168	.171	.19	.209	.228	.247	.285
4 1/4	.115	.144	.168	.192	.216	.24	.264	.288	.312	.36
5	.144	.18	.21	.24	.27	.30	.33	.36	.39	.45
5 1/4	.173	.216	.252	.288	.324	.36	.396	.432	.468	.54
6	.205	.256	.299	.342	.385	.428	.471	.513	.555	.641
6 1/4	.245	.307	.391	.409	.461	.512	.563	.614	.668	.800
7	.279	.348	.408	.466	.524	.583	.641	.699	.756	.874
7 1/4	.321	.401	.468	.534	.602	.669	.735	.802	.869	1.002
8	.365	.456	.532	.608	.685	.761	.837	.912	.989	1.121
8 1/4	.413	.516	.602	.688	.774	.86	.946	1.032	1.118	1.29
9	.462	.577	.674	.770	.866	.963	1.059	1.154	1.251	1.444
9 1/4	.515	.644	.751	.859	.966	1.074	1.181	1.288	1.395	1.610
10	.571	.714	.833	.952	1.071	1.190	1.309	1.428	1.547	1.785
10 1/4	.63	.787	.919	1.050	1.181	1.313	1.444	1.575	1.706	1.969
11	.691	.864	1.008	1.152	1.296	1.44	1.584	1.728	1.872	2.160
11 1/4	.754	.943	1.1	1.257	1.414	1.572	1.729	1.886	2.043	2.357
12	.820	1.025	1.195	1.366	1.540	1.708	1.880	2.050	2.222	2.564
13	.964	1.206	1.407	1.608	1.809	2.01	2.211	2.412	2.613	3.015
14	1.119	1.398	1.631	1.864	2.097	2.331	2.564	2.797	3.029	3.495
15	1.285	1.606	1.873	2.131	2.409	2.677	2.945	3.212	3.479	4.004
16	1.461	1.824	2.131	2.436	2.741	3.045	3.349	3.654	3.958	4.567
17	1.643	2.057	2.396	2.739	3.081	3.424	3.766	4.108	4.450	5.135
18	1.849	2.312	2.697	3.083	3.468	3.854	4.239	4.624	5.009	5.78
19	2.061	2.577	3.006	3.436	3.865	4.295	4.724	5.154	5.583	6.442
20	2.292	2.855	3.331	3.807	4.285	4.759	5.234	5.731	6.186	7.138
21	2.518	3.148	3.672	4.197	4.722	5.247	5.771	6.296	6.800	7.869
22	2.764	3.455	4.031	4.607	5.183	5.759	6.334	6.911	7.486	8.638
23	3.021	3.776	4.404	5.035	5.662	6.294	6.923	7.552	8.181	9.44
24	3.289	4.111	4.797	5.482	6.167	6.853	7.538	8.223	8.908	10.279
25	3.569	4.461	5.105	5.948	6.692	7.436	8.179	8.923	9.566	11.053
26	3.861	4.826	5.630	6.435	7.239	8.044	8.848	9.652	10.456	12.065
27	4.159	5.199	6.066	6.932	7.799	8.666	9.532	10.399	11.265	13.098
28	4.477	5.596	6.529	7.462	8.395	9.328	10.261	11.193	12.125	13.991
29	4.805	6.006	7.007	8.008	9.009	10.01	11.011	12.012	13.013	15.015
30	5.141	6.426	7.497	8.568	9.639	10.71	11.781	12.852	13.923	16.065
31	5.486	6.865	8.001	9.144	10.287	11.43	12.573	13.716	14.866	17.145
32	5.846	7.308	8.526	9.744	10.962	12.18	13.398	14.616	15.834	18.270
33	6.216	7.770	9.065	10.360	11.655	12.959	14.245	15.54	16.835	19.425
34	6.59	8.238	9.611	10.984	12.357	13.73	15.103	16.476	17.849	20.595
35	6.993	8.722	10.199	11.656	13.113	14.57	16.027	17.484	18.941	21.855
36	7.401	9.252	10.794	12.336	13.878	15.42	16.962	18.504	20.046	23.130
37	7.819	9.774	11.403	13.033	14.861	16.29	17.919	19.548	21.177	24.435
38	8.246	10.308	12.026	13.744	15.462	17.18	18.896	20.616	22.334	25.770
39	8.648	10.86	12.67	14.48	16.29	18.1	19.91	21.62	23.53	27.15
40	9.139	11.424	13.328	15.232	17.136	19.04	20.944	22.848	24.752	28.560
41	9.604	12.006	14.007	16.008	18.009	20.00	22.011	24.012	26.013	30.015
42	10.065	12.594	14.693	16.792	18.901	20.99	23.089	25.188	27.287	31.485
43	10.56	13.20	15.4	17.6	19.8	22.0	24.2	26.4	28.6	33.0
44	11.046	13.818	16.121	18.424	20.727	23.03	25.333	27.636	29.939	34.545
45	11.563	14.454	16.863	19.272	21.681	24.09	26.399	28.908	31.317	36.135
46	12.086	15.128	17.666	20.144	22.662	25.18	27.668	30.216	32.754	37.770
47	12.614	15.768	18.396	21.024	23.652	26.28	28.908	31.536	34.164	39.420
48	13.146	16.446	19.187	21.928	24.669	27.41	30.151	32.152	35.633	41.115
49	13.713	17.142	19.999	22.856	25.713	28.57	31.427	34.284	37.141	42.855
50	14.28	17.85	20.825	23.8	26.775	29.75	32.725	35.7	38.675	44.625
51	14.832	18.54	21.665	24.76	27.855	30.95	34.045	37.08	40.205	46.425
52	15.437	19.296	22.512	25.728	28.944	32.16	35.376	38.592	41.868	48.240
53	16.041	20.052	23.394	26.736	30.078	33.42	36.762	40.104	43.446	50.123
54	16.656	20.82	24.29	27.76	31.23	34.7	38.17	41.64	45.11	52.05
55	17.275	21.594	25.193	28.792	32.391	35.99	39.589	43.188	46.787	53.985
56	17.909	22.386	26.117	29.848	33.579	37.31	41.041	44.772	48.503	55.905
57	18.557	23.296	27.062	30.928	34.794	38.66	42.526	46.392	50.258	57.99
58	19.214	24.018	28.021	32.024	36.027	40.03	44.033	48.036	52.039	60.045
59	19.902	24.852	28.924	33.136	37.278	41.42	45.562	49.704	53.846	62.13
60	20.558	25.698	29.981	34.264	38.547	42.83	47.113	51.396	55.679	64.245

The formula shows that for a given horse power ($= H$) and size ($= S \times D$) of engine, the value of $\frac{F_o}{A}$ may be made to vary with the three factors p_m , $\frac{W}{A}$ and w' or N .

The principal quantities which effect the value of p_m are the initial pressure $p_i = AJ$ and the cut-off $\frac{AB}{JK}$; the most important considerations governing the choice of these two quantities will be given a little later. The value $\frac{W}{A}$ cannot become smaller than a certain value $\frac{W^1}{A}$ prescribed by the strength and stiffness of the reciprocating parts. The following limits employed in practice for high speed engines may be of service:

w' varies ordinarily from 600 to 1000 ft. per m.

$\frac{W}{A}$ " " " 2 to 6 lbs.

$\frac{S}{D} =$ " " " 0.8 to 2.

The larger values of w' and the smaller values of $\frac{S}{D}$ are usually employed for large engines. In using the following tables we first assume a diameter and with our given value of $\frac{H}{p_m}$ ($=$ I. H. P. for each pound of mean effective pressure per $\square''$ of piston area) we find from Table IX the corresponding piston speed, or we assume a given piston speed and find from it and $\frac{H}{p_m}$ the corresponding diameter. If we now assume a given ratio of $\frac{S}{D}$ we can get from Table X the number of revolutions N and $\frac{W}{A}$, or if

we assume $\frac{W}{A}$ we can, from the same table, get S and N . It should be noticed that within certain limits we can diminish N and yet increase S without disturbing other established quantities.

V.

PRELIMINARY ESTIMATE OF DIMENSIONS OF RECIPROCATING PARTS.

Calculate the weight of hollow piston head, the proportions being assumed like those given in Reuleaux's *Constructeur*, Fig. 798, p. 746, but with an additional plate at the lower end so that the head will be closed at both ends.

Also estimate the weight of piston rod from the table on p. 751 of Reuleaux's *Constructeur*, remembering that the length of piston rod is greater than the stroke by, length of piston head, length of stuffing box, length of cylinder cover and amount that enters cross-head. The weight of the cross-head in the present design is small, and in this rough estimate may be taken at 35 lbs. The weight of the connecting rod may also be got at roughly by supposing it to be of uniform rectangular cross-section throughout, its dangerous cross-section being calculated from the formulas given at the end of book. Assume the ratio of depth of connecting rod to its width to be $\frac{h}{b} = 2$. Adding together the weight of piston head, piston rod and the connecting rod, and then dividing the sum by the area of the piston we get the quantity $\frac{W}{A}$ which must be equal or less than the quantity $\frac{W}{A}$ prescribed, by the chosen dimensions and by the speeds or value $\frac{F_o}{A}$.

TABLE X.

$$\text{VALUES OF } \frac{W}{A} + \frac{F_o}{A} = \frac{W}{F_o} = \frac{1}{.0000142SN^2}$$

TO FIND $\frac{W}{A}$ FROM TABLE MULTIPLY BY ASSUMED VALUE OF $\frac{F_o}{A}$

$\frac{W}{A}$ = weight of reciprocating parts per $\square''$ of piston.

S = length of stroke in inches. N = revolution per minute.

F_o = average accelerating force per $\square''$ of piston.

QUANTITIES BRACKETED IN TABLE ARE LENGTH OF STROKE = S ,

FOR DIFFERENT VALUES OF N AND PISTON SPEED $\left(w^2 = \frac{2NS}{12}\right)$.

Piston speed in ft. per minute.	Revolutions per minute = N .									
	100	150	200	250	300	350	400	450	500	600
600	(36.0) .1955	(24.0) .1303	(18.0) .0978	(14.4) .0782	(12.0) .0652	(10.29) .0559	(9.0) .0489	(8.0) .0435	(7.2) .0391	(6.0) .0366
650	(39.0) .1788	(26.0) .1186	(19.5) .0889	(15.6) .0712	(13.0) .0593	(11.14) .0508	(9.75) .0444	(8.67) .0395	(7.8) .0356	(6.5) .0297
700	(42.0) .1676	(28.0) .1118	(21.0) .0838	(16.8) .0670	(14.0) .0559	(12.0) .0479	(10.5) .0419	(9.33) .0372	(8.4) .0335	(7.0) .0281
725	(43.5) .1618	(29.0) .1078	(21.75) .0809	(17.4) .0647	(14.5) .0539	(12.43) .0462	(10.88) .0405	(9.66) .0360	(8.7) .0324	(7.25) .0270
750	(45.0) .1564	(30.0) .1067	(22.5) .0782	(18.0) .0626	(15.0) .0521	(12.76) .0447	(11.25) .0391	(10.0) .0348	(9.0) .0313	(7.5) .0261
775	(46.5) .1510	(31.0) .1010	(23.25) .0767	(18.6) .0606	(15.5) .0505	(13.37) .0433	(11.63) .0379	(10.33) .0337	(9.3) .0301	(7.75) .0252
800	(48.0) .1466	(32.0) .0981	(24.0) .0733	(19.2) .0587	(16.0) .0489	(13.71) .0419	(12.0) .0369	(11.66) .0326	(9.6) .0293	(8.0) .0244
825	(49.50) .1422	(33.0) .0948	(24.75) .0710	(19.8) .0570	(16.5) .0474	(14.4) .0406	(12.38) .0356	(11.0) .0316	(9.9) .0284	(8.25) .0237
850	(51.0) .1380	(34.0) .0920	(25.5) .0690	(20.4) .0553	(17.0) .0460	(14.57) .0394	(12.75) .0345	(11.33) .0307	(10.2) .0276	(8.5) .0231
875	(52.5) .1341	(35.0) .0893	(26.25) .0670	(21.0) .0536	(17.5) .0447	(14.71) .0383	(13.13) .0335	(11.66) .0298	(10.5) .0268	(8.75) .0223
900	(54.0) .1303	(36.0) .0868	(27.0) .0652	(21.6) .0521	(18.0) .0434	(15.43) .0377	(13.5) .0326	(12.0) .0290	(10.8) .0261	(9.0) .0217
950	(57.0) .1235	(38.0) .0823	(28.5) .0618	(22.8) .0506	(19.0) .0412	(16.29) .0359	(14.25) .0309	(12.66) .0274	(11.4) .0247	(9.5) .0211
1000	(60.0) .1172	(40.0) .0782	(30.0) .0587	(24.0) .0469	(20.0) .0381	(17.14) .0335	(15.0) .0293	(13.33) .0261	(12.0) .0235	(10.0) .0196
1050	(63.0) .1117	(42.0) .0745	(31.5) .0553	(25.2) .0447	(21.0) .0373	(18.0) .0319	(15.75) .0280	(14.0) .0241	(12.6) .0224	(10.5) .0186
1100	(66.0) .1066	(44.0) .0711	(33.0) .0533	(26.4) .0427	(22.0) .0356	(18.86) .0305	(16.5) .0267	(14.66) .0237	(13.2) .0213	(11.0) .0182
1150	(69.0) .1020	(46.0) .0680	(34.5) .0510	(27.6) .0412	(23.0) .0340	(19.71) .0292	(17.25) .0254	(15.33) .0221	(13.8) .0204	(11.5) .0174
1200	(72.0) .0976	(48.0) .0652	(36.0) .0489	(28.8) .0391	(24.0) .0326	(20.57) .0279	(18.0) .0244	(16.0) .0217	(14.4) .0196	(12.0) .0163

VI.

DETERMINATION OF WEIGHT OF FLY-WHEEL RIM.*

Let m_o represent the mass of the fly-wheel, w_o its weight, V_1 its maximum, V_2 its minimum, and $V = \frac{V_1 + V_2}{2}$ its average speed of rim in feet per second. Then will maximum variation of energy equal

$$\frac{m_o}{2} (V_1^2 - V_2^2) = \frac{m_o}{2} (V_1 + V_2)(V_1 - V_2) \quad (51)$$

the coefficient of unsteadiness is

$$f = \frac{\overline{V_1} - \overline{V_2}}{V} \quad (52)$$

hence variation of energy =

$$f m V^2 = f \frac{w_o}{g} V^2 = .000085 f D_o^2 N^2 w_o \quad (53)$$

D_o being diameter of rim in feet. N = Revolutions per minute. From the phase of greatest variation of tangential pressure diagrams we get $\overline{Mm}$ and $\overline{MON}$ then if A = area of piston in $\square''$ and R' = length of crank in feet we have variation of energy =

$$\overline{Mm} \times A \times \frac{\overline{MON}}{360} \text{ or } 2\pi R' = .000085 f D_o^2 N^2 w_o \quad (54)$$

$$w_o = 205 \frac{A \times \overline{MON} \times \overline{Mm} \times R'}{f D_o^2 N^2} \quad (55)$$

According to *Der Taschenbuch des Ingenieurs* the coefficient of unsteadiness $f = \frac{V_1 - V_2}{V}$ where $V = \frac{V_1 + V_2}{2}$ varies as follows.

* The figure and formulas of this article were taken from Rigg's Treatise on the Steam Engine.

For machines which will permit a very uneven motion, hammers, etc.,

$$f = \frac{1}{5}$$

For machines which permit some irregularity, pumps, shearing machines, etc.,

$$f = \frac{1}{20} \text{ to } \frac{1}{30}$$

For machines which require approximation to uniform speed, as in flour mills,

$$f = \frac{1}{25} \text{ to } \frac{1}{35}$$

For machines with tolerably uniform speed as weaving and paper making.

$$f = \frac{1}{30} \text{ to } \frac{1}{40}$$

For cotton spinning machinery requiring very uniform speed.

$$f = \frac{1}{40} \text{ to } \frac{1}{60}$$

For the spinning machinery of very high yarn numbers*

$$f = \frac{1}{100}$$

The areas (see figure) included between mean tangential pressure circle and that portion of the tangential pressure curve lying outside of the mean tangential pressure circle we will call phases of excess of power; the areas included between the mean circle and that portion of tangential pressure curve within the circle of average resistance. These phases may be numbered as in figure by Roman numerals, and the angles at the center which they subtend, in degrees. The average excess or deficiency of pressure of any phase may be found tentatively, by drawing arcs subtending the angle of each phase as in figure, the area included between these arcs and the mean tangential circle being equal to the areas of their corresponding phases.

k is the ratio, $\frac{\text{Excess of power or resistance during any phase}}{\text{Total power exerted during a revolution}}$

$$= \frac{\overline{Mm}}{\overline{Aa}} \times \frac{\overline{MON}}{360} = \text{coefficient of variation of energy.}$$

* For electric lighting. $f = \frac{1}{100}$

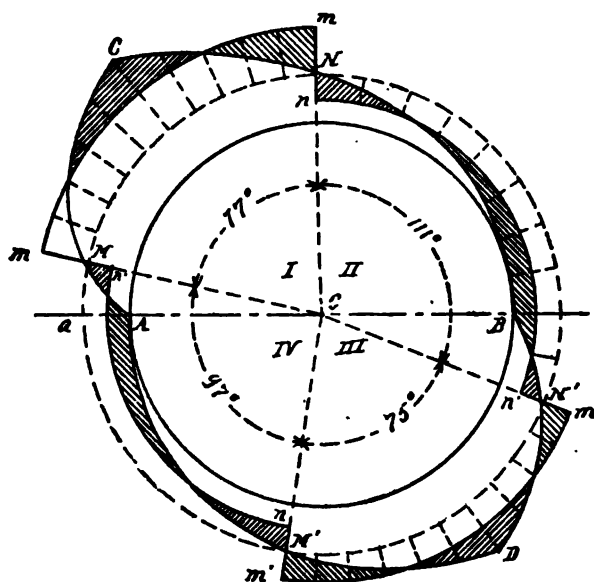


Fig. 33.

$$f = \text{coefficient of unsteadiness} = \frac{V_1 - V_2}{V}{}^2 \text{ when } V = \frac{V_1 + V_2}{2}$$

$$= \text{mean velocity of rim. } f = \frac{1}{50} \text{ in present case.}$$

A = area of piston in $\square''$.

R = length of crank in inches.

Mm = excess of pressure in lbs. (see figure).

MON = angle (in degrees) of phase of greatest variation.

D_o = diameter of fly wheel in feet.

N = revolutions per minute.

H' = indicated horse-power of engine when the greatest variations of power or resistance occur.

$\bar{V}$ = velocity of rim per second ≤ 80 feet. Assume = 64 feet in present case.

W_o = weight of fly wheel rim. (It may be diminished 8% for arms and boss.)

$$W_o = 17.1 \frac{AR \overline{Mm} \overline{MON}}{fD_o^2 N^2} \quad (56)$$

or

$$W_o = 388000000 \frac{kH'}{fD_o^2 N^3} \quad (57)$$

If in the figure on the preceding page $\overline{Mm} = 2.9$ $Nn = 1.9$, $\overline{N'm} = 1.9$, $\overline{M'n} = 1.6$ and $\overline{Aa} = p_t = 3.1$, the areas of the various phases will be measured by the following products

I	Excess of Power	$= 2.9 \times 77^\circ = 223.3$	
II	" " Resistance		$1.9 \times 111^\circ = 210.9$
III	" " Power	$= 1.9 \times 75^\circ = 142.5$	
IV	" " Resistance		$1.6 \times 97^\circ = 155.2$
		Excess of Power = 365.8	Excess of Resistance = 366.1

Had the work been perfectly accurate the two results would have been exactly equal. It is evident that in the present example I is the phase of greatest variation, consequently its average ordinate $\overline{Mm}$ and angle $\overline{MON}$ should be substituted in formula for weight of fly wheel rim. The ratio which the measure 223.3 of phase I bears to the measure $\overline{Aa} \times 360 = 3.1 \times 360 = 1116$ of the total work done in one revolution is $\frac{223.3}{1116} = 0.22 = k$.

Find from the three tangential pressure diagrams for H , H_1 , H_2 the phase of greatest variation and substitute in formula for fly-wheel:

In a manner similar to that detailed on preceding page we can find the phase of greatest variation from the tangential pressure diagrams drawn on the rectified crank pin circle. Instead of multiplying the average pressure of each phase by the size of the phase expressed in degrees, we multiply by the distance between the two intersections of each phase with the mean tangential pressure line.

VII.

BALANCING THE RECIPROCATING PARTS BY COUNTER-WEIGHT
ON CRANK.

We have already seen that only a portion P' of the effective steam pressure P is transmitted to the crank pin during the first portion of the stroke, the remainder ($P - P'$) being engaged in accelerating the reciprocating parts. The absorption of pressure by the moving parts disturbs the statical equilibrium of the forces acting on the engine bed $BCDA$. The steam pressure P acting on the rigid frame at A being greater than P' the horizontal force at B , the tendency is to shift the engine bed in the direction CD

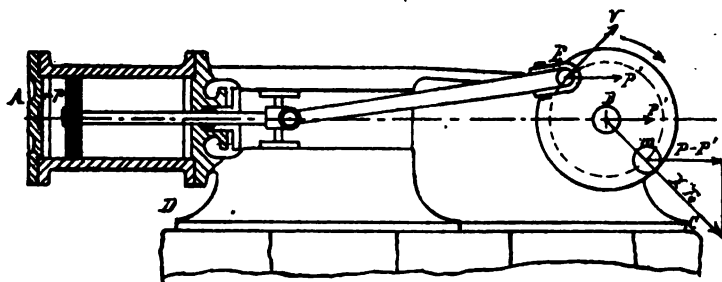


Fig. 34.

of the greater force P , the shifting force being equal to $P - P'$. Similar considerations will show that as the piston approaches the end of its stroke this shifting force changes in direction and tends to slide the engine bed on its foundations in the opposite direction DC . In order that this effect may be wholly or partially neutralized by other means than the employment of numerous bolts for holding the engine bed to its foundations, a counter-weight m or its equivalent is attached to the crank as shown in the figure. The horizontal component $P - P'$ of the centrifugal force xF_0 developed by this mass m when revolving in the crank pin circle with velocity V will then be transmitted to the point B of the engine bed and together with P' will make a force equal to P and will balance the steam pressure P acting at the other end A of the frame.

The horizontal component of the centrifugal force is easily obtained from the expression

$$x F_2 \cos \omega = m \frac{v^2}{R} \cos \omega = \frac{w}{g} \frac{v^2}{R} \cos \omega \quad (58)$$

where m represents the mass engaged in balancing the reciprocating parts reduced to a point on the crank pin circle diametrically opposite to the crank pin, v = velocity of crank pin, R = radius of crank, ω = crank angle, we have already shown that the force F accelerating or retarding the reciprocating parts is given by the expression

$$F = \frac{W}{g} \frac{v^2}{R} \left(\cos \omega + \frac{R}{L} \cos 2\omega \right) \quad (59)$$

where W represents the weight of the reciprocating parts. The frame will be in statical equilibrium and will have no tendency whatever to shift on its foundations

$$\text{when} \quad \frac{w}{g} \frac{v^2}{R} \cos \omega = \frac{W}{g} \frac{v^2}{R} \left(\cos \omega + \frac{R}{L} \cos 2\omega \right) \quad (60)$$

$$\text{that is when} \quad w = \frac{W \left(\cos \omega + \frac{R}{L} \cos 2\omega \right)}{\cos \omega} \quad (61)$$

It is evident from this equation that equilibrium cannot exist for all values of ω unless the counter-weight w also varies with the crank angle ω . This it is of course not practicable to do, hence there will always be a shifting force acting on frame equal in amount to

$$\begin{aligned} & \frac{W}{g} \frac{v^2}{R} \left(\cos \omega + \frac{R}{L} \cos 2\omega \right) - \frac{w}{g} \frac{v^2}{R} \cos \omega \\ &= F_o \left(\cos \omega + \frac{R}{L} \cos 2\omega - x \cos \omega \right) \\ &= F_o \left[(1 - x) \cos \omega + \frac{R}{L} \cos 2\omega \right] \quad (62) \\ & \quad x \text{ being equal to } \frac{w}{W} \end{aligned}$$

The maximum shifting effort when reciprocating parts are not balanced by counter-weight corresponds to $\omega = 0$ and $x = 0$; hence maximum shifting force equals

$$F_o \left(1 + \frac{R}{L} \right)$$

and when the reciprocating parts are balanced the maximum shifting force corresponds again to $\omega = 0$ and is equal to

$$F_o \left(1 - x + \frac{R}{L} \right).$$

When $x = \text{unity}$, that is when counter-weight $m = W = \text{weight of reciprocating parts}$, the ratio of these two maximum shifting forces is

$$\frac{R}{L} \div \left(1 + \frac{R}{L} \right).$$

It will generally suffice to make $x = .5$ to $.8$; the unbalanced portion being resisted by the foundations.

Calculate the value of the vertical components of the mass m .

Instead of employing the counter-weight as shown in the preceding figure it is customary to employ a crank disk of the following shape.

The crank proper is balanced by a counter-weight of similar shape shown in dotted lines, there remain therefore only the two portions $ABCDE$ and $A'B'C'D'E'$ having the depth w which can be utilized as counter-weight for balancing the reciprocating parts, the dotted lines $A'F'G'E'$ and $AFGE$ representing the counter-balance for part (if any) of the weight of connecting rod, which part is supposed to be concentrated at crank pin. It is evident from the figure that the center of gravity c of the portions $ABCDEF$ and $A'B'C'D'E'F'$ does not fall upon the crank pin circle, consequently this deficiency of radius must be made good by increasing the masses $ABCDEF$ and $A'B'C'D'E'G'F'$. Let the sum

of these larger masses be represented by M' , its radius by r , then we must have

$$\frac{M'(v')^2}{r} = \frac{m'r^2}{R}$$

and since $\frac{v'}{v} = \frac{r}{R}$ we have $M' = \frac{R}{r} m'$. (63)

The value of r can be found experimentally by suspending the figure $ABCDEGF$ from two of its corners and determining the

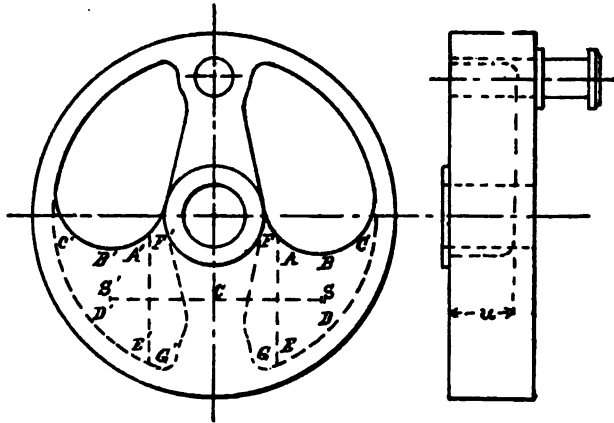


Fig. 35.

intersection of the two lines of suspension, or it may be balanced over a knife edge in two different positions. M' can then be computed.

The dimensions of the simple crank can be calculated from the formula given by Reuleaux.

To find room on the crank disk for the large mass M' of counterweight, the outside circumference of crank disk, must be taken considerably larger than crank pin circle. By casting hollows in the disk and filling them with lead, all the counter-weight desired can be placed on crank disk. When double cranks or disks are employed the disposal of counter-weight becomes an easy matter.

VIII.

EFFECT OF FRICTIONAL RESISTANCES, OF THE WEIGHT OF THE ROD AND EXACT VALUES OF THE FORCES OF INERTIA, ON THE ROTATIVE EFFORT, ON THE PIN PRESSURES AND ON THE FORCE SHAKING THE ENGINE BED.

The effect of these three influences on the rotative effort and energy of the fly-wheel has been very fully considered by Prof. D. S. Jacobus* in two papers, published in Vol. XI, Transaction of American Society of Mechanical Engineers.

In these papers very accurate formulas were established which took account of these three influences. They were applied to four different engines representing a wide range of practice. The exact diagrams constructed for these engines showed that the fluctuation of energy, ΔE , of the fly-wheel due to the variation of the rotative effort differed by an insignificant amount from the approximate diagrams found by supposing all the reciprocating parts (inclusive of rod) concentrated on piston. A similar result was obtained for the crank-pin pressures, the nearly exact ones differing but slightly in direction and intensity from those found by assuming the mass of the rod to be all concentrated at wrist-pin. The following tables are given by Prof. Jacobus and furnish the data of the four engines and the results of the numerical computations.

* Prof. Jacobus credits Prof. J. B. Webb with valuable assistance in the preparation of these two papers.

TABLE XI.
DIMENSIONS AND SPEED OF ENGINES TO WHICH THE FORMULAS
HAVE BEEN APPLIED.

	Class of Engine.			
	I	II	III	IV
	Small horizontal high-speed.	Large horizontal high-speed. Locomotive.	Large horizontal slow-speed of revolution. Harris— Corliss.	Westinghouse.
Revolutions of crank shaft per minute . . .	300	250	60	320
Length of stroke, in inches	12	24	60	10
Diameter of cylinder, in inches	10	18½	26½	11
Length of connecting-rod, in inches	36	92	150	5 R
Distance from the wrist-pin to the center of gravity of rod, in inches	20.15	55	78	3.22 R
Distance from the center of the crank shaft to the line of travel of the wrist-pin, in inches	0	0	0	0.5 R
Principal radius of gyration, in inches . .	15.00	34.1	48	2.07 R
Weight of piston, piston-rod, and cross-head, in pounds	90.	474	1300	100
Weight of connecting-rod, in pounds . . .	70.	307	1200	50
Indicated horse-power	57	345*	216	66

* For one cylinder.

TABLE XII.

RESULTS OF NUMERICAL APPLICATIONS TO SEVERAL STANDARD CLASSES OF ENGINES.

Class of Engine.	Conditions assumed.	$\frac{\Delta E}{\int Pds}$			Pressure acting on crank-pin.		
		Exact.	Approximate.	Not including accelerating forces.	Exact.	Approximate.	Not including accelerating forces.
I	Not including the effects of friction and weight	.184	.174	.238	58.1	57.3	83.0
I	Not including friction but including weight . . .	.180	. . .	. . .	58.0	. . .	. . .
I	Including both friction and weight	.171	. . .	. . .	56.8	. . .	. . .
II	Not including friction and weight	.181	.172	.275	56.8	55.5	108.0
III	Not including friction and weight	.142	.131	.185	76.1	75.5	89.4
IV	Not including friction and weight	.160	.147	.247	61.0	60.8	75.5

The term approximate in these tables refers to the assumption that the whole mass of the rod is concentrated at wrist-pin. The expression $\frac{\Delta E}{\int Pds}$ is the ratio of the periodical excess or deficiency of energy ΔE , to the whole energy exerted per revolution, $\int Pds$. The particular crank-pin pressure given in table is the maximum one and is in pounds per square inch of piston area.

It is evident from these tables that for most practical work the aforesaid approximation, used in the preceding sections, is sufficiently accurate. The remainder of this section may therefore be omitted by those who do not care to examine the refinements of this subject.

We have not reproduced here any of Prof. Jacobus' formulas for the components of the forces acting on the pins, though they are easily applied, and, after the constants have been computed and introduced, lead quickly to the desired results. We have preferred graphical determinations because they are simple, and naturally accompany designing. We will therefore follow graphical methods in finding exact values for each of the three influences under consideration.

The order in which we will take them up is:

- a.* The friction between a pin and its bearing.
- b.* The direction of internal stress in a rod, when friction is taken into account, but gravity and inertia neglected.
- c.* The direction of internal stress in a rod, when friction and some other force, say, resultant of gravity and inertia-resistance, are considered.
- d.* Determination of the exact accelerating force of the rod corresponding to its motion in the slider-crank chain.
- e.* Combination of weight of rod with its force of inertia.
- f.* The components of this total force at wrist- and crank-pin, when friction is neglected.
- g.* The components of this force at these pins, when friction is considered.
- h.* Determination of the force shaking the engine bed.
- i.* Diagrams of shaking forces with different degrees of counterweighting.
- j.* Diagram of pressures at crank- and at wrist-pin.

A.

THE FRICTION BETWEEN A PIN AND ITS BEARING.

The frictional resistances of motion and rest differ by quantities that are directly proportional to the coefficients of friction for motion and rest, respectively. By using the coefficient of friction for motion, we may treat the body as if it were at rest and yet

determine the direction and intensity of the forces as they exist under running conditions.

Let us suppose the pin to be at rest, without tendency to move in either direction, and that the resultant pressure P_o between the pin and its bearing acts in the direction $c a e f$, Fig. 36. Then the pin can be brought to the eve of, say, left-handed rotation, in one of two ways, either by the action of an additional, single, force Q , with lever arm $c l$, or by the action of some couple M having left-hand rotation.

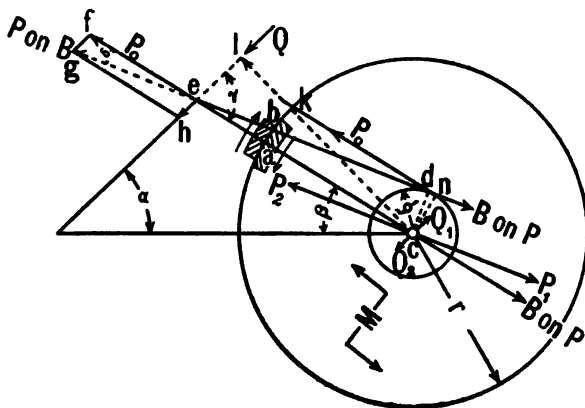


Fig. 36.

In the first case there will be an infinite number of solutions depending upon the location and direction of the turning force Q . The moment $F'r$ of the friction of the bearing will balance the moment $Q \times \bar{cl}$ of the new force and the point of application of the resultant pressure between pin and bearing will be shifted from a to b , sufficient play being supposed to exist between the pin and its bearing to permit this to take place. The point b is where the resultant $P (=eg)$ of the forces $Q (=ch)$ and $P_o (=ef)$ cuts the area of contact. Evidently the new resultant P is different in intensity and direction from the original pressure P_o . The pin is kept stationary by $+P$ and $-P$; turning of the pin is prevented because

$$Q \times \bar{cl} = P \times \bar{cd} = Pp' = F'r = \varphi.Nr.$$

The resultant pressure P between bearing and pin is therefore necessarily a tangent to the friction circle whose radius $\rho' = \varphi r$ when P is taken equal to N . This is permissible because the angle of friction is so small, that we may regard its sine and tangent as equal.

The force P for this first case can also be constructed by placing at center c two equal and opposite forces $Q_2 = Q_1 = Q$. Then Q and Q_1 will form a couple which, combined with the original force P_o , will shift the latter parallel to itself through a distance $\frac{P}{P_o} \rho'$; now combining P_o in this new position with the force Q_2 remaining at the center we get the same force P as before.

In the second case the pin is brought to the eve of left-hand rotation by the action of some left-hand couple M which may act anywhere in the plane. This case does not often arise in practice, and is chiefly useful in demonstrations involving equivalence of forces. The effect of this couple is to shift the force P_o parallel to itself to a position in which it is tangent to the so-called "friction circle" having radius $\rho' = \varphi r$. The point of contact for this case and the direction and intensity of its force P_o are all different from those obtained for the first case. As the couple M may act anywhere in the plane, let us suppose it replaced by an equivalent couple $Q_o \times \bar{cl}$ whose two equal and opposite forces Q_o act respectively at the center C and along the line of turning force Q mentioned in the first case. Then it is easy to show that this force Q_o is different from Q . In Fig. 36, we have given, the angle γ , the radii r and ρ' and the lever arm cl . Then

$$\sin \delta = \frac{\bar{cd}}{\bar{cc}} = \frac{\bar{cd} \sin \gamma}{\bar{cl}} = \frac{\rho' \sin \gamma}{\bar{cl}} \text{ and } \frac{P}{P_o} = \frac{\sin \gamma}{\sin (\gamma - \delta)};$$

but

$$Q \times \bar{cl} = P \times \rho'$$

$$\text{and } Q_o \times \bar{cl} = P_o \times \rho' \text{ then } \frac{Q_o}{Q} = \frac{P_o}{P} = \frac{\sin (\gamma - \delta)}{\sin \gamma}, \quad (64)$$

$$\frac{Q_o}{P_o} = \frac{Q}{P} = \frac{\rho'}{cl} = \frac{\varphi r}{cl}. \quad (65)$$

As each piece is subjected to at least two forces, the driving force and the resistance, there will always be a force Q available for bringing the pin or shaft to the eve of turning. We will therefore assume the first case as the common one. It should be noticed that the resultant action of bearing on pin is tangent to that side of friction circle which will give this resultant a component opposed to the rotation of the pin or shaft. In like manner the action of pin on bearing is such a tangent to the friction circle, that it has a component opposed to the bearing's (relative) rotation. This is because friction is a hindrance to motion. In Fig. 36, these equal and opposite actions are indicated by arrows and the letters P on B and B on P .

B.

THE DIRECTION OF INTERNAL STRESS IN A ROD ACTED UPON BY TWO PINS.

There are four cases under this head, which are due to the two kinds of relative motion possible between a pin and its bearing, and to the two kinds of stress, tension or compression, to which the rod may be subjected. The four figures* on next page illustrate these four cases.

We will suppose the link AB , Fig. 37, in all cases to carry the eyes or bearings into which fit the pins belonging to the cranks MA and NB . The large arrows about the center N or M indicate which crank is the driver and the small arrows about the pin centers A and B indicate whether the rotation of rod AB to crank MA or NB is right- or left-handed. The arrows on the rod AB itself indicate whether it is in tension or compression.

* These figures were taken from Herrmann-Smith's Graphical Statics of Mechanisms.

In cases I and II the relative motions of the pins A and B to their bearings in the rod AB are alike, being both right-hand rotations, but the internal stress is different in the two cases, being tension in case I and compression in case II.

In cases III and IV the relative motions of pins B to their bearings in the rod are still right-handed, but those of the pins A to their bearings are now left-handed. The inner stresses are also different in these two cases, tension in case III and compression in case IV.

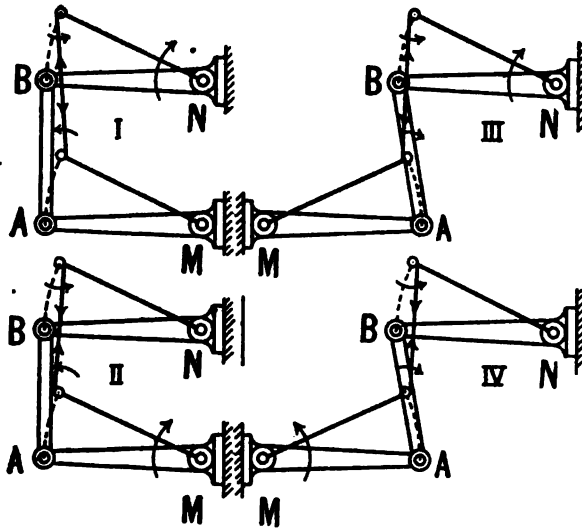


Fig. 37.

When there is no friction the inner stress acts along the center line of the rod in each case.

When there is friction, the direction of the inner stress is different in each of these four cases.

In Figs. 38–41 sections are taken of both pin and its bearing near the point of contact and arrows added to indicate the relative motion of the rubbing surfaces.

Now that there is friction, we do not at first know the point of application of the resultant of the forces acting at each surface of

contact; but we do know that the action ap , Fig. 38, of pin A on bearing must have a component opposed to the (relative) motion of the bearing. This action ap prolonged backward will therefore be tangent to lower portion of friction circle cde . In

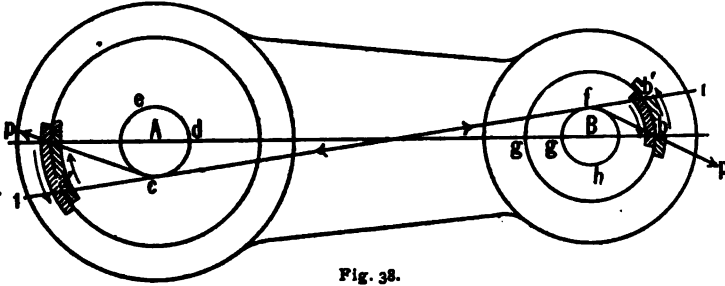


Fig. 38.

like manner the action of pin B on its bearing when prolonged backward will also be tangent to its friction circle fgh , but at the upper portion of the circle. Neither ap nor bp represent the two directions of the forces exerted by the pins, for friction has changed the intensities of these forces and shifted their points of application to a' and b' . As there are but two forces acting on the rod, it can only be in equilibrium when the two forces are directly opposite and equal. This condition determines the

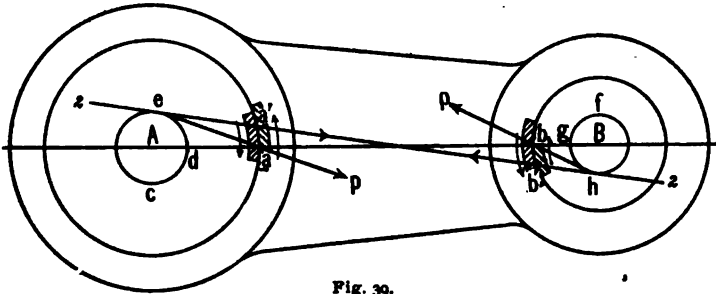


Fig. 39.

direction of the inner stress; it must be tangent to each of the friction circles of A and B as in the figure, the pin A acting at a' and pin B at point of application b' .

In the case shown in Fig 39 the rotation of the pins is still right-handed, but the rod is now subject to compression and con-

tact takes places at a and b on sides of the pins opposite to those of Fig. 38. Similar reasoning shows that in this case the direction of the inner stress must be along the line $2ek2$.

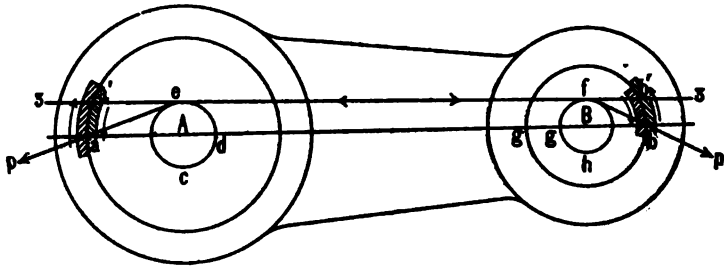


Fig. 40.

In Fig. 40 pin B keeps its right-hand rotation, but pin A has left-hand rotation, the rod being in tension. The figure indicates the condition of equilibrium and direction $3ef3$ of the inner stress.

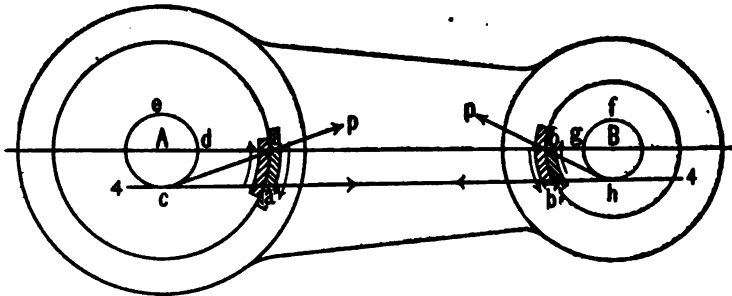


Fig. 41.

In this last case, Fig. 41, pin B has still right-hand rotation and A left-hand rotation. Compression however now exists in the rod and this brings the contacts a and b different from the preceding case, namely, on the opposite sides of the pins A and B . The tangent ch on line 44 is the direction of the inner stress.

These four cases have furnished four lines of stress coinciding with the four possible tangents, 11, 22, 33 and 44 to the friction circles. This would also have been the case, if the rod had carried the pins instead of the eyes or bearings.

C.

DIRECTION OF INTERNAL STRESS IN ROD WHEN ITS ACCELERATING FORCE IS KNOWN.

The known accelerating force KG , Fig. 42, is the resultant of the unknown forces acting at the crank-pin C and wrist-pin W . When there is no friction we will designate these forces by P_c and P_w respectively and their directions must be assumed to pass through their respective centers C and W . When there is friction the forces will be designated by P_{cf} and P_{wf} and they must then be tangent to their respective friction circles. In either case, unless some additional condition is given with reference to one or the other of these forces, the value of P_c or P_{cf} , P_w or P_{wf} is indeterminate for there are evidently in each of these cases an infinite number of pairs of forces which can produce the given accelerating force KG .

The forces P_{wf} and P_{cf} (or P_c and P_w) evidently produce both acceleration $PH (= KG)$ and whatever internal stress may exist. This latter will vary according to the pair of forces assumed to produce the acceleration and also according to the points of application assumed for the members of this pair. For instance, let us assume that PF and PE represent the directions of the pair of forces P_{cf} and P_{wf} , then any points F' , F , S and E' , E , T in the directions of these forces may be assumed as the points of application of the forces, and the corresponding directions of the internal stress will be ST , FE , $F'E'$. The intensities of these internal stresses along either of these directions will depend entirely upon the acceleration components of the forces P_{wf} and P_{cf} . These components acting at the assumed points of application S , T or F , E or F' , E' have only one condition to fulfill, namely, to produce the acceleration PH . As this can be accomplished in an infinite number of ways, at the same pair of points of application, it is evident that those other components of P_{wf} and P_{cf} , the internal stresses, have also an infinite number of

$Sm = PM = P_{wf}$ and $Tn = MH = P_{cf}$, we get Sd and Tq as the acceleration components and Sb , Ta the components producing the internal stresses in the rod along the line TS . Measurement will show that $Sb = Ta$. In like manner we might have assumed the acceleration components Sd' and Tq' both parallel to the total acceleration KG ; then resolving P_{wf} and P_{cf} we get $Sd' + Tq' = KG$ and internal stress $Sb' =$ internal stress Ta' . If F and E had been chosen as the points of application, FE would have been line of internal stress. Then if FV and EV had been chosen as direction of acceleration components, $Fe = Eh$ would represent the equal internal stresses. It is evident that there is great freedom in the choice of the direction of the internal forces and later on we will indicate how this may be so chosen as to lead very directly to the determination of the actual pressures exerted by the pins upon their bearings in the connecting rod. In the engine problem these pressures are not indeterminate as they are here, for there the known effective steam pressure and the direction of the reaction of the guides, together furnish another condition, which is all that is necessary to completely determine P_{wf} .

D.

EXACT DETERMINATION OF FORCE NEEDED TO ACCELERATE ROD.

Any motion of a body in a plane is equivalent to the rotation of the body about its instantaneous axis. The rotation about this axis is, by a principle of Kinematics equivalent to an equal rotation about any other, second, axis *plus* a translation, the latter motion being equal to that possessed by the second axis when rotating around the first, or instantaneous, axis. This equivalence of motion extends not only to the displacements, but also to the velocities and accelerations. The axis passing through the center of gravity will be taken as this second axis because it is the most convenient one, the centripetal accelerations about this axis completely neutralizing each other.

The proposition may also be stated in this better known form:

The total accelerating force of a body, whose points move in parallel planes, is compounded of the force needed to give the whole mass an acceleration of translation equal to that of its center of gravity and of a moment (or couple) capable of imparting to the body an angular velocity and acceleration about its own center of gravity equal to that possessed by the body when rotating about the instantaneous axis.

The combination of this force of translation with the moment or couple, gives a single resultant force that is equal and parallel to its component force of translation but has a location that is different from this component, the effect of the couple being merely to shift the location of the force producing the translation, a couple with $\left\{ \begin{array}{l} \text{left-hand} \\ \text{right-hand} \end{array} \right\}$ rotation shifting the force to the $\left\{ \begin{array}{l} \text{left} \\ \text{right} \end{array} \right\}$; in applying this rule the plane of forces must be viewed so that the force of translation is downward.

The moment of the force or couple capable of producing the desired angular acceleration $\frac{d\Omega}{dt}$, is, according to Mechanics,

$$\mathbf{I} \frac{d\Omega}{dt} = m k^2 \frac{d\Omega}{dt} \quad (66)$$

where $\mathbf{I}$ is the principal, polar, moment of inertia, m the mass of the connecting rod and k the corresponding radius of gyration.*

This expression is perfectly general so far as the distribution of the mass of the rod is concerned. Any arrangement of this mass which will not alter its center of gravity and which preserves the same total moment of inertia $\mathbf{I}$ unchanged will require the same total accelerating force as the rod itself. The problem may consequently be greatly simplified for graphical purposes by supposing the mass of the rod concentrated at two points that are on opposite sides of the center of gravity and in line with it.

* For graphical and experimental methods of determining this polar moment of inertia, see Appendix.

In order that the equivalence of total accelerating force may be maintained it is necessary that the following three conditions be fulfilled,

$$m = m_1 + m_2, \quad (67)$$

$$m_1 h_1 = m_2 h_2 \quad (68)$$

$$mk^2 = m_1 h_1^2 + m_2 h_2^2, \quad (69)$$

and by combining these we get

$$k^2 = h_1 h_2, \quad (70)$$

where h_1 and h_2 are the distances from the center of gravity of the two points at which the mass is supposed to be concentrated and m_1, m_2 the masses at these points. Equations (67) and (68) keep the total mass and location of the center of gravity the same as before and therefore keep unchanged the accelerating, component, force due to the translation of the center of gravity. The fulfillment of equation (69) or (70) makes the turning moment of the rearranged, two-point, rod the same as that of the original rod with distributed mass. As the force and energy of this two-point rod is the same as the original one, we can confine our determination of the total accelerating force to the simplified, two-point rod.

The problem is now reduced to finding the acceleration of the center of gravity of the rod and that of each of its two points of concentration. The problem is therefore now a purely kinematic one and is a special case of the general problem of finding the acceleration of any point on the connecting rod of the slider-crank mechanism,

In this general case of the mechanism and its motion, the acceleration CO , Fig. 43, of the crank-pin center C is known and the acceleration $Ww'' (= Ow)$ of the wrist-pin center or slide can be obtained as in Fig. 16 and p. 43. The accelerations of two points of the rod will then be known and the center of acceleration G of the rod can be found.* The acceleration of any point of the

* See Weisbach-Hermann's Machinery of Transmission, Vol. III, Section I, § 21.

rod is directly proportional to the distance of this point from the center of acceleration G , and, for the instant in question, the acceleration of each point makes the same angle with its own

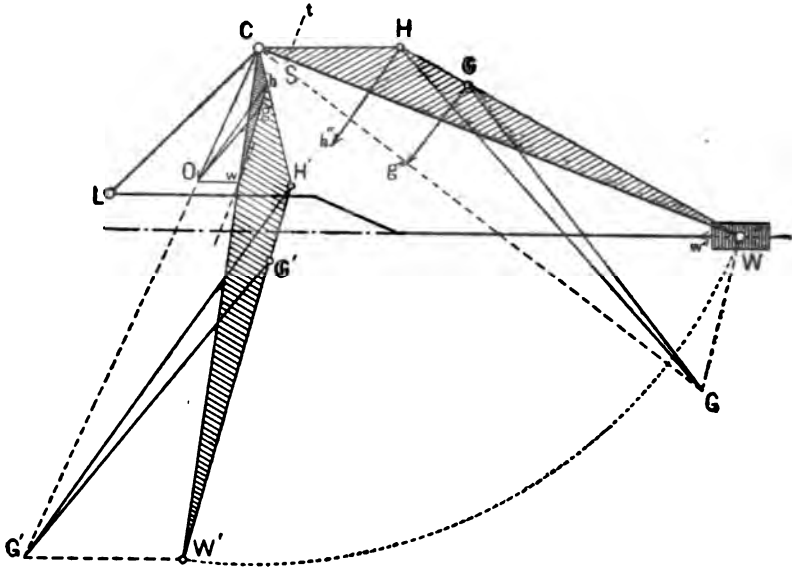


Fig. 43.

instantaneous radius of acceleration.* This solves the problem but not in a convenient way because the center of acceleration generally falls beyond the limits of the drawing. A method that is free from this objection and also simpler in other respects is found as follows:

In Fig. 43 revolve the triangle CWG (having the rod CW as a base and acceleration-center G as a vertex) through an angle GCG' , equal to that made by the direction of acceleration of point C with its instantaneous radius of acceleration CG . Then will the revolved radii $G'C$, $G'\mathcal{G}$, $G'H'$ and $G'W'$ be parallel to the directions of the accelerations of points C , $\mathcal{G}$, H and W , as well as proportional to the magnitude CO , $\mathcal{G}g''$, Hh'' and Ww'' of

* See last paragraph on p. 45 and first one on p. 46 for one method of finding acceleration of any point of rod.

these accelerations. As the triangle COw is similar to the triangle CGW and $Chgw$ to the rod $CH\odot W$, the vectors OC , Oh , Og and Ow drawn from O as a pole will represent in direction and magnitude the accelerations of the points C , H , $\odot$ and W of the rod.

The pole O is the end of the acceleration CO of crank-pin and when Ow is found by method given in Fig. 16 and p. 43, the figure $Cwgh$ can easily be made similar to the rod $CW\odot H$. Since H may be any point in the rod we have a simple method of finding its acceleration (without first finding the center of acceleration) provided the direction and intensity of the acceleration of two points of the rod are known.*

We can now give two simple constructions for the total force accelerating the rod of the ordinary slider-crank, when crank has uniform rotation.

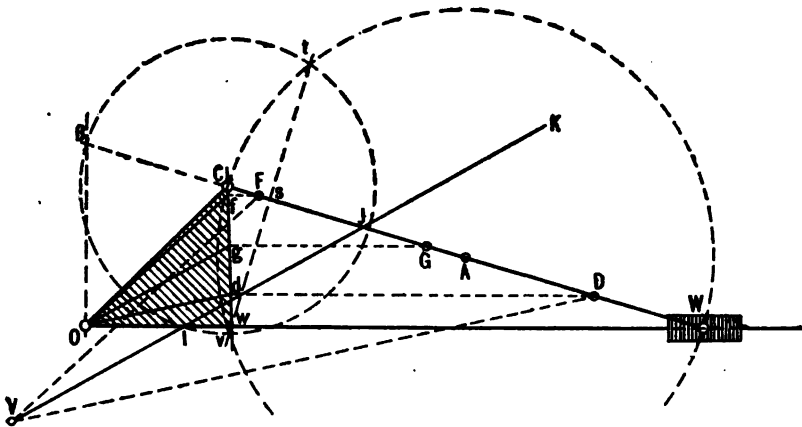


Fig. 44.—First Construction.

The mechanism OCW , Fig. 44, is the ordinary slider crank, in which the stroke of the slide W passes through the center O of the crank OC . The center of the rod is at A , the center of gravity at G and $GF = GD$ is the principal polar radius of gyration.

* This same simple method may also be deduced from a more general case. See Journal Franklin Institute for September and October 1891,

We first construct the acceleration of the slide W by the method given on p. 43 and Fig. 16.

Prolong the rod WC till it intersects at B the perpendicular OB to the stroke OW . With C as a center and CB as a radius describe a circle Btv . On the rod CW as a diameter describe another circle. Join the intersections of these circles by the chord tsv and prolong it, if necessary, till it cuts a line drawn through crank center O and parallel to stroke. In Figs. 44 and 45 this intersection is at w , and OW is the desired acceleration of the slide, provided crank length CO represents the centripetal acceleration of the uniformly rotating crank-pin.

(The construction would be exactly the same for the crossed slider crank of Fig. 46 if its crank had uniform rotation. This construction of the slide acceleration has no failing case even at the dead point.)

Now join C and w . Cw is a reduced image of the rod, making with it an angle equal to that made by each point's acceleration with its own radius of acceleration and point O has the same positions relatively to the image Cw of the rod that the center of acceleration has relatively to the rod itself. It follows from this that if we draw through points F , G and D of the rod, parallels Ff , Gg and Dd to the stroke, that the distances or vectors Of , Og and Od will represent in direction and intensity the accelerations of F , G and D , respectively. The actual locations of their accelerations will, of course, be parallels through these points to those vectors. One-half of the mass of the rod is supposed to be concentrated at F and the other half at D .

Draw FV parallel to Of and DV parallel to Od . Through their intersection V draw, parallel to Og , the line VK ; it will be the actual location of the resultant force of inertia, and its intensity will equal the product of the whole mass of the rod by acceleration $\overline{Og}$ of the center of gravity.

In Fig. 45 the mass m of the rod is divided into two unequal parts, one of which m_1 is concentrated at W and the other m_2 at

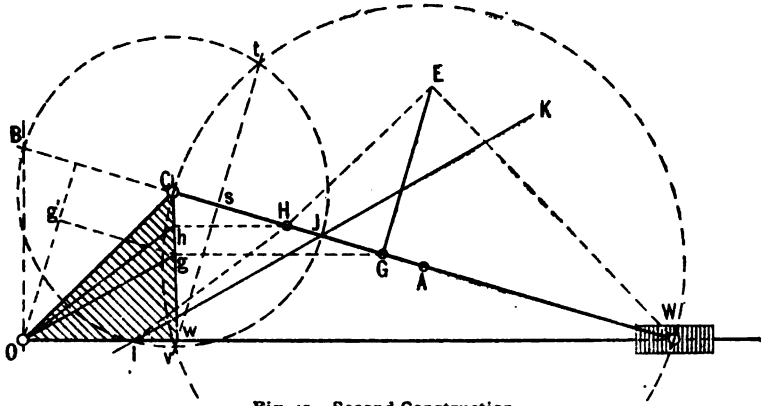


Fig. 45.—Second Construction.

H , the relation between the points W , G and H being $\overline{GW} \times \overline{GH} = k^2$, where k is the principal polar radius of gyration. The two masses must be to each other as

$$m_1 : m_2 = \overline{GH} : \overline{GW}.$$

Construct as before the acceleration Ow of the slide and Cw the image of the rod. At G erect the perpendicular $GE = k$. Join E with W and draw EH at right angles to EW . This will evidently satisfy the condition $\overline{GW} \times \overline{GH} = k^2$.

Through H and G draw Hh and Gg parallel to the stroke, then will the vectors Oh and Og represent in direction and intensity the accelerations of points H and G , respectively.

Through H draw HI parallel to Oh ; it will cut at I the direction WI of the acceleration of the other mass at W . Through I draw, parallel to Og , the line IK ; it will be the actual location of the resultant force of inertia of the rod and its intensity = whole mass of rod $\times$ acceleration Og of center of gravity.

These two constructions give, of course, exactly the same result, the first being a little the easiest when the intersection V falls within the limits of the paper. The great advantage of the second is that the direction of W is given by the line of stroke, and that the intersection I will always fall within the limits of the

drawing. Both constructions of the total force of inertia fail at the dead point, but this is of no consequence, because then the direction and location of the total acceleration of the rod is known, for it coincides with the stroke, and, as before, its intensity = the whole mass $\times$ acceleration Og of the center of gravity.*

This construction is only one of an infinite number of possible cases, each of which satisfies the three conditions involved in equations (67), (68) and (69) or (70).

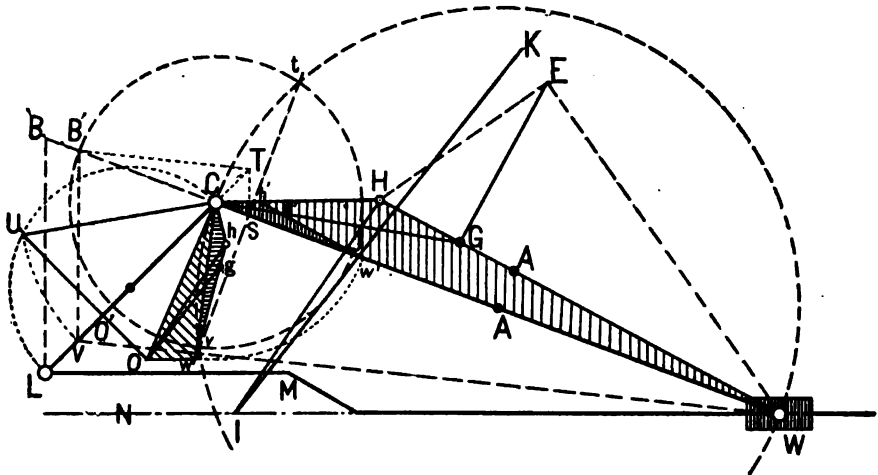


Fig. 46.

It is evident that $h_1 = h_2$ is a particular case of the general one just given, and is illustrated in Fig. 44.

Fig. 46 is meant to be the most general case of slider-crank, with

* The nearer point H , Fig. 45, is to crank-pin C the more nearly does the point of intersection I preserve a constant position. In the rods of good high-speed-engines the locus of point I varies but little, the average value of $\overline{OI}$ being nearly equal to $CH = L - \frac{h_1^2 + k^2}{h_1}$; where $L = CW$ is length of connecting rod and $h_1 = GW$, Fig. 45, is distance of center of gravity G from wrist-pin W . This could be made the basis of a close approximation, in which the point I is treated as a fixed point and IK is drawn at once parallel to $\overline{Og}$, thus omitting the drawing of the lines Hh , hO and HI .

respect to the mechanism itself, to its motion and to the distribution of the mass of the rod. The mechanism LCW is of the crossed slider-crank type in which the stroke of slide W does not pass through center L of crank. The rotation of crank CL is not uniform as in the preceding cases, the acceleration of crank-pin C , taking (say) the direction and intensity CO instead of CL as before. The center of gravity of the mass is no longer in the center line CW of the rod, but outside of it, at G . The point H is on the line WG as before, and again satisfies the condition $GW \times GH = k^2$, where k is the principal polar radius of gyration. H is found by the same construction, namely, we erect the perpendicular $GE = k$, join EW , and draw EH at right angles to EW .

The velocity of the crank-pin is no longer represented by CL , as in Figs. 44 and 45, but by CV . It is found by construction, as follows: On crank CL as a diameter construct a circle LUC ; from end O of acceleration draw a perpendicular $OO'U$ to CL ; then will $CU = CV$ be the intensity of the velocity of the crank-pin C . To find the acceleration Ow of slide, we proceed as in Figs. 44 and 45, except that a circle is described with CB' as a radius instead of CB .*

In Fig. 46 the distance OC and Ow , respectively represent the

* There is still another way of finding point S (and consequently point w) that is always available.

Draw VB' perpendicular to stroke WN and prolong it till it cuts the direction WC of the rod at B' . Join W and V , make $B'T$ parallel to WV and draw TS perpendicular to WN .

For the similar triangles $B'CV$ and SCT give

$$CS : CT :: CB' : CV \text{ or } CB' \times CT = CS \times CV,$$

and the similar triangles $B'CT$ and WCV give

$$CB' \times CV = CT \times CW.$$

Multiplying these two equations and canceling, we get,

$$CS = \frac{CB'^2}{CW}$$

and this is similar to the result found for BF' , Fig. 16, pp. 41-43.

accelerations of crank-pin C and wrist-pin W . The line Cw is the reduced image of the center line CW of the rod. We may consider this rod to be an exaggerated specimen of the Westinghouse type in that its center of gravity G is off the center line CW of the rod. We may regard the triangle $CHGW$ as the representation of the rod and proceed to construct a reduced image of it on Cw as a base. To do this lay off $Cw' = Cw$, draw $w'h'$ parallel to WH and connect C with G . The triangle $Ch'g'w'$ is evidently similar to $CHGW$. Revolving it back to Cw as a base we get triangle $Chgw$ as the desired image of the rod, and Oh , Og will be, in direction and intensity, the accelerations of points H , G , respectively. Through H draw HI parallel to Oh ; it will cut at I the direction WI of the acceleration of the other mass at W . Through I draw, parallel to Og , the line IK ; it will be the actual location of the resultant force of inertia of the rod, and its intensity, as before, is equal to the product of the whole mass of rod $\times$ acceleration Og of the center of gravity.

COMBINATION OF WEIGHT OF THE ROD WITH ITS FORCE OF INERTIA.

An exact determination of the pressure exerted by rod against the crank- and wrist-pins must include its weight.

We will, therefore, find the resultant of the weight and of the resistance due to inertia. As the inertia is measured by mass times acceleration, the expression of the force of inertia becomes:

$$\text{Inertia of rod} = \frac{W}{g} \times \frac{\overline{Og}}{\overline{OC}} \times \frac{v^2}{R} = F_2 \quad (71)$$

W = weight of rod in pounds;

v = velocity of crank-pin in feet per second;

R = radius of crank in feet; and

g = acceleration due to gravity = 32.16 feet.

Og , OC = accelerations of center of gravity and of crank-pin as given by Figs. 44 and 45. In the case of Fig. 46 we must

Exact computation* gives

$$\frac{F_2}{A} = 4.37.$$

which differs but 0.01 pounds per square inch from the result obtained by the graphical method. Considering the small, scale, one-twelfth, employed, this shows the practical excellence of the graphical method.

In the above example the weight is one-half the inertia-resistance, but that is because the engine is large and of comparatively slow speed. In small, high-speed engines, the proportion is much less. Thus, in a *N. Y. S. S. P.* engine, 10 x 12, with rod of seventy pounds and 300 r. p. m. (*i. e.*, same crank-pin speed as in the large engine) we found

$$\frac{W''}{A} = 0.89 \quad \text{and} \quad \frac{F_2}{A} = 9.21$$

or weight = one-tenth inertia. In the latter case the resultant will differ but little from the inertia in direction and intensity.

F.

PRESSURES AT CRANK- AND WRIST-PIN WHEN THERE IS NO FRICTION AT THE PIN.

When there is no friction at the pins, let P_c and P_w represent respectively the crank-pin and wrist-pin pressure. Let line lq , Fig. 48, represent the direction of the resultant of the weight and inertia of rod, and let the intensity of this be assumed equal to $F_3 = lp = kh$. It is evident P_c and P_w combined must equal this resultant, and if one of these forces (say P_w) is known, the other P_c can be found by the parallelogram of forces. To obtain P_w we must satisfy two conditions, one of which is that P_w must do its

* The computation was made by the help of formulas developed by Profs. Jacobus and Webb, in a paper published in *Trans. Am. Soc. Mech'l. Eng.*, vol. xi. The data for the above example was obtained from the same source (see p. 498).

part in producing the resultant F_3 acting along lq , and the other that P_w must balance the resultant of the guide reaction G and the driving force K of the piston. This force K is the difference between the effective steam-pressure P and the inertia F of piston, piston-rod and cross-head.

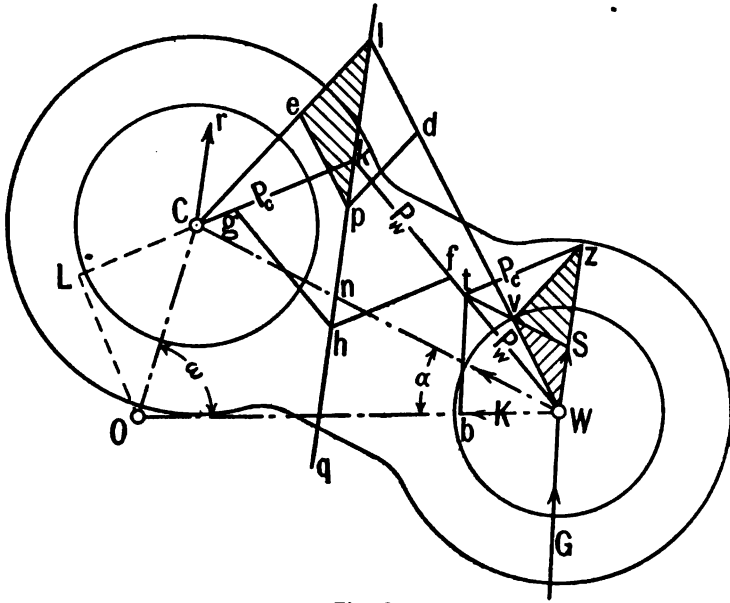


Fig. 48.

Through W draw Wz equal and parallel to $F_3 = lp = kh$. Join any point l on lq with C and W . Draw zv and Wv , respectively parallel to lC and lW . Through the intersection v draw vs parallel to center line CW of rod, which line also represents direction of internal stress, as C and W are the points of application of the forces. We will show later that this parallel ts is the locus of the ends of all forces applied at W that satisfy the first condition mentioned above. Now lay off $Wb = K = P - F_i =$ driving force and draw bt parallel to guide reaction G . The line bt is the locus of the ends of all forces applied at W that satisfy the second condition. The intersection t of these two loci

St and bt , gives $Wt = P_w$, the force desired. Prolong Wt to point k on line of resultant, then making $kh = F_3$ and $kf = Wt = P_w$ and completing the parallelogram, we get the crank-pin pressure $P_c = kg$ and the rotative effect $= P_c \times OL$.

For convenience of expression we will speak of the two components of each pin pressure as internal-stress component and inertia-component. *The latter expression is not strictly correct as the second component not only overcomes the inertia but also sustains the weight of the rod*, but this will not at all affect the accuracy of our results.

Since the force P_c passes through crank-pin center C and may have its point of application anywhere on its line of direction without affecting its accelerating capacity, we will assume this point to be at C , and since force P_w passes through wrist-pin center W , we will, for the same reason, take its point of application at W . This will make the center line CW of the rod the direction of the internal stress caused by either P_c or P_w . Although the intensity of this internal stress varies as the inertia components P_c and P_w are varied, nevertheless there is but one pair of forces P_c , P_w that will satisfy the conditions of this problem. To prove this it is only necessary to show that there is only one force P_w that will satisfy the conditions at the wrist-pin. One condition is that it shall be in equilibrium with the known force $K = P - F_1 = Wb$ and the partly known reaction G of the guides. By drawing through b a parallel bt to the known direction of G , we get one locus for the end of the force P_w . Another condition that P_w must satisfy is, that it shall be the resultant of the internal stress acting along CW and the corresponding inertia component acting in any arbitrarily chosen direction. Let Wl be any such direction, then will Cl be the direction of the corresponding inertia component that acts at the crank-pin C . If $kh = lp = Wz = F_3$ is the known resultant of the weight and inertia of the rod, then $ld = Wv$ is the intensity of the inertia component acting along the assumed direction Wl , and $le = vs$

is the inertia component along Cl . Now draw through v a parallel Svt to the inner stress CW ; it will divide the triangle Wvs into two triangles WSv and svS , respectively similar to triangles Wnl and lnC . Hence

$$\overline{nW} \times \overline{SW} = \overline{vS} \times \overline{ln} = \overline{nC} \times \overline{Sz}$$

and

$$\frac{WS}{Sz} = \frac{nC}{nW} = \text{constant} \quad (72)$$

for this position and motion of the rod. Since Wz is also constant for this position of rod, the parallel Svt will pass through the same point S whatever the direction and intensity Wv of the inertia component. Therefore, the intersection of Svt with the locus bt will give Wt as the only possible value of P_w for this position and motion of the rod.

Prolonging P_w to k and drawing kC and also tz , we get $P_c = kg = tz$. It is also evident from the figure that the reaction $G = bt$ of the guide is independent of the direction and intensity of the inertia-component.

G.

PRESSURES AT CRANK- AND WRIST-PINS WHEN THERE IS FRICTION AT THE PINS.

When there is friction at the pins, let P_{cf} and P_{wf} represent, respectively, the crank-pin and wrist-pin pressures, and let GKL , Fig. 49, represent as before the location of the resultant F_3 of weight and inertia of rod. We will again suppose $F_3 = GN = KQ = PO = TL$ to be known. In this, as in the former case, P_{cf} and P_{wf} combined must be able to sustain the weight and overcome the inertia of the rod.

Moreover, P_{cf} must be tangent to a circle described from C as a center with φr_c as a radius, and P_{wf} must be tangent to a similar circle about W with φr_w as a radius when φ is coefficient of journal friction, and r_c, r_w the radii, respectively, of crank-pin

known that P_w passed through the center W of wrist-pin, and that point could, therefore, be assumed as its point of application. In the present case the possible values of P_{wf} do not pass through such a common point, and we must, therefore, make use of trial methods to find P_{wf} .

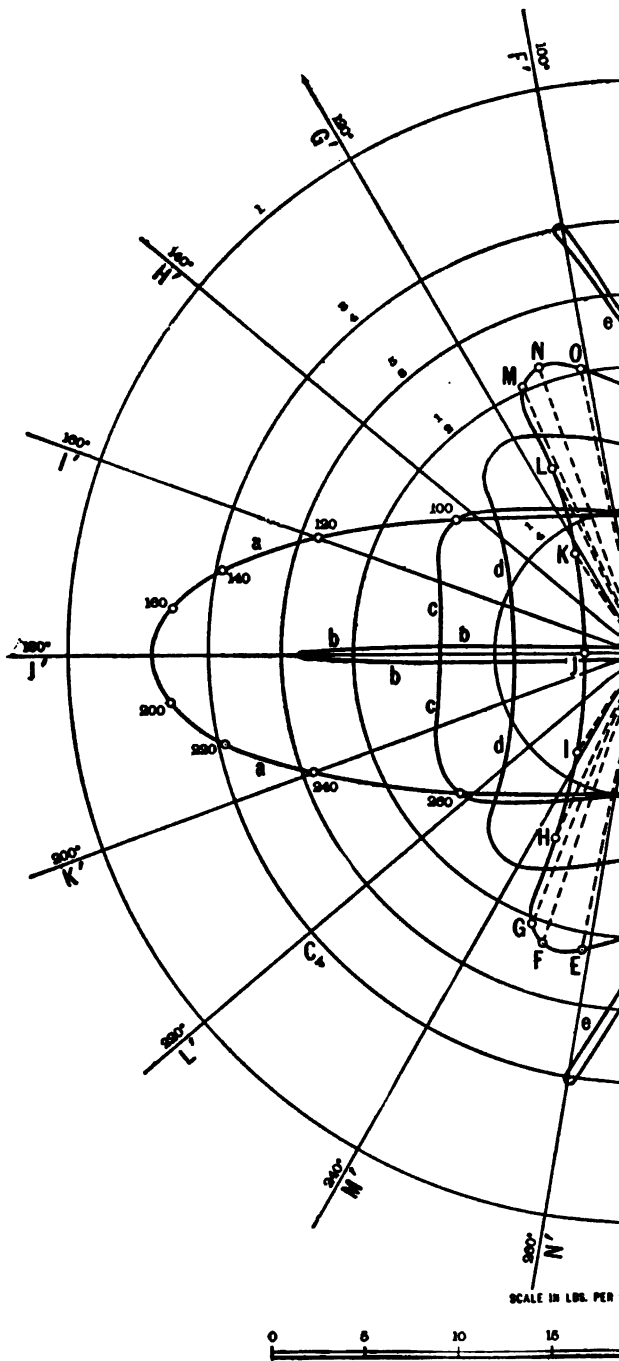
We will first give a method which is, theoretically, the most exact, and then another, which is simpler, and, at the same time, accurate beyond the needs of practice.

From each of the series of points G, P, T , on the line GL draw a tangent to each of the two friction circles mng and ktp . Make distances $GN = PQ = TL$ equal to the known force F_3 and resolve each of these distances into components along the respective tangents. Then with W as a pole draw Wa equal and parallel to GA , Wb equal to PB , also Wd equal and parallel to TD , and so on, thus getting a polar curve $eabd$, whose vectors satisfy the condition that the pin forces shall be capable of balancing the resultant F_3 . The second condition is the same in this as in the preceding case, namely, that the wrist-pin pressure P_{wf} shall be in equilibrium with the driving force $K = \overline{WV} = P - F_1$ and the given reaction G (known in direction only). Now laying off $\overline{WV} = K$, drawing Vb parallel to the guide reaction G' , and joining its intersection b (with the polar curve $eabd$) to the pole W , we get in Wb the exact intensity and direction of the wrist-pin pressure P_{wf} , but not its location. The latter is found by drawing HBP parallel to Wb and tangent to the friction circle plk . Finally, drawing $PB'I$ tangent to friction circle mng of pin C , and constructing the parallelogram of forces, we have $PB = Wb = P_{wf}$ and $PB' = P_{cf}$, the exact pin forces desired. The rotative force is measured by $P_{cf} \times \overline{vw}$.

In the simpler and only theoretically less exact method, the principal steps are like those taken in finding P_c and P_w in Fig. 48. The direction of the internal stress is, however, no longer CW , and the determination of a convenient direction for it, constitutes the main portion of the problem.

To illustrate the method, suppose the problem solved and the forces P_{cf} and P_{wf} just found to be each resolved into two components, one parallel to GL and passing through C and W , respectively, and the other in the direction of the internal stress. The two components parallel to GL must together just equal F_3 and, therefore, be capable of sustaining the weight and overcoming the inertia of the rod. The direction of each of the two internal stress components will evidently be HI , and according to d'Alembert's principle they will just balance each other. Inspection of the figure will show that for friction-circle-tangents differing but slightly in direction from P_{cf} and P_{wf} , the direction of the internal stress will also differ but slightly from HI . This is the case even in the present figure, where the rod is very short and the friction circles excessively large. Now draw P'_w parallel to P_w and tangent to the friction circle plk , the location of this tangent will differ but little from the actual force P_{wf} . For the same reason P'_c parallel to P_c and tangent to qmn , will differ but little from P_{cf} . Joining the intersections F and E (of P'_w and P'_c with the parallel components WM and Ct , respectively), we get in FE a line whose direction is nearly parallel to the exact direction of the internal stress HI .

Our procedure in finding P_{wf} is, therefore, as follows: We get the component WS as it was got in Fig. 48: we next draw through S a parallel Sb to the line FE , just described; then will Sb be a locus of the end b of the desired force Wb . But another locus is needed to completely determine b ; this is given by the condition that $Wb = P_{wf}$ shall be the resultant of the driving force $K = P - F_1$ and guide reaction G' . Therefore, laying off $WV = K$ and drawing Vb parallel to G' we get a second locus whose intersection b with the first locus Sb gives the desired intensity Wb of the force P_{wf} , for Wb is common to each of the force triangles WSb and WVb , thus satisfying both conditions imposed upon P_{wf} . The exact location of P_{wf} may now be found by drawing pBP parallel to Wb and tangent to the friction circle plk .



of the lines representing
no counterweight.
equal to $\frac{1}{4}$ the mass of the
mass of the reciprocating
the mass of the reciprocating
to $\frac{3}{4}$ the mass of the recip-
entire mass of the recipro-
nute, 300.

II.

rod and reciprocating parts
 $\square''$ of piston; hence cen-
s equivalent to $\frac{1}{8}$ ths of the
per $\square''$.

a, a , are given in the last
 b, b, b, d, d, d, d , etc., see

TABLES OF RESULTANT OF RECIPROCATING PARTS.

Sum of the Horizontal Components of Inertia of Moving Parts.	Sum of the Vertical Com- ponents of In- ertia of Con- necting Rod. Lbs. per $\square''$.
35.17	0.00
34.46	1.33
32.37	2.62
29.02	3.83
24.63	4.92
19.42	5.86
13.66	6.63
7.64	7.19
1.70	7.53
—3.99	7.65
—9.18	7.53
—13.72	7.19
—17.57	6.63
—20.74	5.86
—23.21	4.92
—25.06	3.83
—26.34	2.62
—27.06	1.33
—27.29	0.00

The accuracy of the work may be checked by completing the parallelogram $OBPB'$ and seeing if $PB = Wb$.

The approximation just given can be still further simplified for all cases that may arise in practice. In such cases the friction circles are very small and far apart, and the tangent RJ to both friction circles is scarcely distinguishable from the direction HI or EF . We may therefore treat this tangent RJ as the direction of the internal stress, draw Sb parallel and then proceed as above in the determination of Wb and location of its equal P_{wf} . The inaccuracy involved in this last, approximate, construction is much less than that connected with the best determined coefficients of friction.

H.

DETERMINATION OF THE FORCES TENDING TO SHAKE THE ENGINE BED.

The constant forces acting on the engine bed, like the weight of fly-wheel and pull of belt, tend only to shift the bed. It is the variable forces that tend to shake it, namely, the effective steam pressure P against cylinder covers, the pressure of the cross-head against the guides and the pressure of the crank-shaft against its bearings. By constructing a polygon of forces, it can be shown that the force tending to shake the engine bed is the resultant of all the unbalanced accelerating forces acting on the different links of the machine. In other words, the shaking force is the resultant of the forces of inertia of the moving parts.

The same conclusion is reached from general considerations. Suppose the forces of inertia to be replaced by their resultants acting as external forces. Then the whole system of external forces will be in equilibrium, the moving pieces may be considered at rest for the instant, the whole machine acting as one rigid piece. As the steam then acts equally in opposite directions upon the machine, its shifting and shaking influence are both nil. The internal forces at the bearings occur in pairs of equal and

opposite forces, and their shaking influence is therefore likewise nil. The only external forces remaining are the resultants due to inertia, the action of gravity and the pull of the belt. The weights are constant forces, but their points of application change with the motion. In stationary engines we may neglect the shaking influence due to this cause. As regards the pull of the belt Mr. W. Willis has shown by his experiments that the sum of the two tensions is not a constant quantity, as is generally assumed, but increases with the load. If we suppose the load constant the belt pull will also be constant, and will only tend to shift the bed without shaking it. The only external forces remaining are the variable forces of inertia, and these do tend to shake the engine frame. In the case of the engine, therefore, the shaking forces are, the resistances due to the inertia of the purely reciprocating pieces (piston, piston-rod and cross-head), to the inertia of the connecting rod and to the inertia of the crank disc. The first and last are easily obtained by computation, and the second can be obtained by the construction shown in Fig. 44 or Fig. 45.

I.

DIAGRAM OF FORCES TENDING TO SHAKE ENGINE BED.

Fig. 50 is such a diagram prepared by Prof. D. S. Jacobus and published in the Trans. Amer. Soc. Mech'l Eng'rs, Vol. XI. It contains a series of polar curves, *aaa*, *bbb*, *ccc*, etc., whose vectors, drawn from the pole or center *V*, represent in direction and intensity the resultant forces of inertia of all moving pieces. The data for the construction of the diagram are given on the same page, and under Case I, Table XI. The table accompanying Fig. 50 contains the components or the force of inertia for this particular engine, and is in convenient shape for use. The last column contains the ordinates, like C_3C_2 , of the polar curve *aaaa*, and the last column but one the abscissas, like VC_3 , of this same polar curve. As this polar curve *aaaa*, corresponds to the case of no counterweight, its vectors represent in direction and inten-

sity the resultant of the forces of all moving pieces, except the crank. The remaining curves *bbbb*, *cccc*, *dddd*, etc., can therefore be directly deduced from curve *aaaa*, by laying off from each point *a*, and parallel to the crank position corresponding to this point *a* the centrifugal force of counterweight on crank disc.

For instance, when crank is at 40° , the vector VC_2 will be the resultant of the inertia-forces of all the reciprocating parts. To get a point on curve *bbbb*, for the same crank position of 40° , we lay off $C_2b = \frac{1}{4} \times 31.25 = 7.81$ parallel, but opposite to VC' . To get a point on curve *dddd*, for this crank angle of 40° , we lay off on the same line $C_2bd = \frac{5}{8} \times 31.25 = 19.53$ lbs. per $\square''$, and so on for the other curves.

As the vertical components of curve *bbbb* are very small, its small amount of counterweight would be suitable for cases in which the shake occurs more readily in a vertical than in a horizontal direction, as when a horizontal engine is placed on an upper floor of a building. On the other hand the horizontal components of curve *eeee*, are small and the corresponding, heavier, counterweight is preferable when the horizontal forces produce most shake, as in engines set on tall foundations.

The diagram just described is suitable for stationary engines, and can readily be constructed for such cases as soon as the acceleration of the center of gravity of rod has been found graphically, (see page 101, Eq. 71 and Fig. 44 or 45). We have then to multiply this acceleration by the mass of the rod per $\square''$ of piston, and resolve this force into two components, one horizontal and the other vertical. Of these two components add that which is parallel to stroke to the accelerating force (per $\square''$ of piston) of the piston, piston-rod and cross-head. This sum will be one of the coordinates of the curve *aaaa* (Fig. 50). The remaining component of the rod will be the other coordinate of curve *aaaa*. The influence of counterweight can then be added in the manner already described in connection with Fig. 50. The diagram represents the shaking forces acting on engine bed in the vertical and axial plane of the engine.

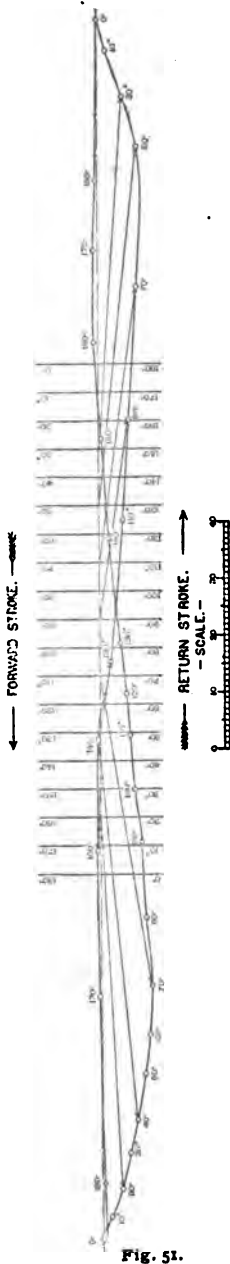


Fig. 51.

When the engine frame is suspended on springs, as in locomotives, the exact location of the total accelerating force of rod should be taken into account. The exact location of the weights is also of consequence in determining the oscillations of the locomotive. Indeed, in this case it would be well to combine separately the force of inertia and weight of each moving piece and then combine these resultants into a total resultant with its point of application on the line of stroke and lay off each of these resultants from its own point of application and with its own direction and intensity.

Construct the curve *aaaa* for the engine to be designed, determining the inertia of the connecting rod by the constructions given in Fig. 44 or 45. For method of ascertaining the radius of gyration experimentally or by computation, see Appendix.

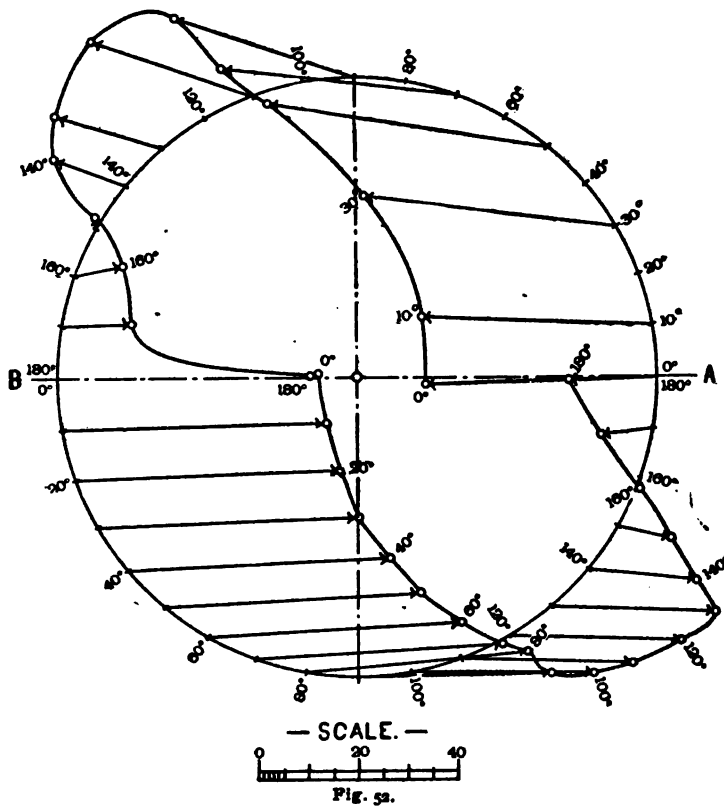
J.

DIAGRAMS OF PRESSURES AT CRANK- AND WRIST-PIN.

Rapid changes in the directions of the forces acting at the pins are approximately indicated by such diagrams as Fig. 27. They do not necessarily indicate reversals, this term being understood to mean an instantaneous change of pressure from one side to the other of a pin. If the pin had been worn into an oval shape then it is possible that the changes shown in Fig. 27 might be accompanied by shocks.

In Fig. 51, the pressure of connecting-rod on wrist-pin is shown in direction and intensity for the engine whose data are given under Case I, Table XI, when friction, weight and inertia are all taken into account. These forces are laid off from what is equivalent to a time base, namely, the developed crank-pin circle. Rapid changes of direction of pressure occur between 140° and 150° for both forward and return strokes. It is not however instantaneous, continuous contact being preserved between the pin and its bearing.

But between slide and its guides the contact very suddenly changes from one side to the other in the neighborhood of 150° forward stroke and 160° return stroke, provided the upward,



vertical, component or pressure of wrist-pin is sufficient to overcome the weight of cross-head and that part of piston rod which is sustained by the slide. In such a case a pressure of slide against guide would suddenly change from a downward to an upward pressure and be accompanied by a more or less violent shock.

In Fig. 52 the pressure of connecting-rod on crank-pin is given in direction and intensity for the same engine and conditions assumed in Fig. 51. Between 140° and 160° of forward stroke and 150° and 170° of return stroke a rapid change of direction of force occurs, but continuous contact between pin and eye is maintained even then and no shock will occur at crank-pin under these particular circumstances. At the inner, or front dead point *A* there is a sudden change of pressure on crank-pin, which may cause tremor in the rod but no shock.

IX.

DIAMETER AND WIDTH OF BELT PULLEY.

Generally in high speed engines the fly wheel acts also as belt pulley. In such a case the diameter is determined by the data given on pp. 75 and 76. When the diameter is not fixed in this way or by some special condition and there is thus left some choice, it should be so made as to give the belt a high speed, even a mile a minute is permissible, for the belt becomes narrower and the efficiency increases with the speed, the percentage of slip and the journal friction being smaller at the higher speed.*

* See pp. 347-360 also pp. 568 of Vol. VII, Trans. Amer. Soc. M. E. Mr. W. Lewis in experiments made for Sellers & Co., found that with vertical belts, the sum of tensions while running might increase to $\frac{1}{4}$ of the sum existing when belt is stationary; that with horizontal belts the sum of the tensions might increase up to the breaking limit. In both cases this increment in the sum is accompanied by increase of slip. In one extreme case, with the permissible slip of 3 feet per minute, the increment of the sum was as high as 100%. This does not call for any modification of the results given above however, as the coefficient of friction is dependent upon the slip and the assumption of a 0.27 coefficient means a definite slip and a ratio of tensions, in tight and slack sides, of about 2.4.

When belt speed is known, its width can be ascertained from the following table, provided the belt is laced and connects two equal pulleys,

Belt speed = 1000, 2000, 3000, 4000, 5000, in ft. per min.									
Horse-Powers transmitted	}	=	5.0	9.8	14.3	18.3	20.8	}	per □" of belt.

This table assumes a permissible working stress of 300 lbs. to the square inch at laced joint and a coefficient of friction = 0.27 for leather on cast iron. For a riveted joint the working stress would be larger say from 400 to 550, and the H. P. transmitted per □" of piston from 1.4 to 2 times greater.* Some authorities use a coefficient of friction of 0.40, which would make the H. P. per □" in the above about $\frac{1}{4}$ larger, but it would be accompanied by a velocity of sliding of belt on pulley of much more than three feet per minute.

When pulleys connected by belt are unequal, account must be taken of the arc of contact on the *smaller* pulley, by multiplying the horse-power given above by the fractions in this next table:

Arc of contact =	120°	130°	140°	150°	160°	170°	180°	degrees.
Multiplier	= 0.79	0.83	0.87	0.91	0.94	0.97	1.00	

The maximum horse-power that the engine is likely to develop must then be divided by the H. P. per □" of the belt found by these tables and the result will be the area of the cross-section of the belt. The area is given because there is considerable variation in belt thickness, and the mere designations of single and double do not necessarily mean $\frac{7}{8}$ " and $\frac{7}{16}$ " thickness respectively. Of course to get the belt width the area of belt is divided by the thickness.

In this engine assume that a double belt is to be used on the fly wheel as a belt pulley and that the arc of contact on the driven pulley is 165°.

Many high speed engines, like the center-crank type, are provided with two fly wheels or belt pulleys and when both are used

for the purposes of transmission there will be a corresponding reduction in the area or width of each belt.

It is usually in connection with Fly or Band Wheels that we must distinguish between right and left hand engines, between those that "run over" and those that "run under." Standing at the cylinder end and looking towards the crank, a $\left\{ \begin{array}{l} \text{right-hand} \\ \text{left-hand} \end{array} \right\}$ will have the fly wheel and main bearing to the $\left\{ \begin{array}{l} \text{right} \\ \text{left} \end{array} \right\}$ of the axis of the cylinder, and still retaining the position at the cylinder, if the top of the fly wheel moves $\left\{ \begin{array}{l} \text{away from} \\ \text{towards} \end{array} \right\}$ the observer, the engine is said to $\left\{ \begin{array}{l} \text{run over} \\ \text{run under} \end{array} \right\}$.

X.

GRAPHICAL DETERMINATION OF DIAMETER OF CRANK SHAFT.

The problem is one of compound stress, and has been treated by Reuleaux in his *Konstrukteur*. Both bending and twisting forces strain the shaft. The bending forces are: weight of the fly-wheel, pull of the belt, the centrifugal force of the counter-weight on crank disc, the pressure of connecting rod against crank-pin and the reaction of the bearings. The twisting force is the moment of variable pressure of connecting rod against crank-pin.

As the bending forces are not parallel, and do not all act in the same plane, the first step is to combine these separate bending moments into one resultant bending moment. The next step is to combine this resultant bending moment with the twisting moment so as to form either an equivalent bending, or an equivalent twisting moment. In the present case we will reduce all the moments to one equivalent bending moment.

The first step may be simplified by certain preliminary reductions and approximations. The weight of fly-wheel and the pull of the belt act in the same plane, and can be reduced to a single

force. The belt pull is the sum of the tensions on the tight and slack side of the belt, and this sum is not constant, but increases, while running, with the load. But this does not call for special calculation at this place, because this increase in the sums of the tensions is limited by the slip, and this was assumed at 3 feet per min. as a maximum. We will therefore assume the sum of the tension to be $1\frac{1}{2}$ times the tension on the tight side of the belt when transmitting the maximum power. This, under our assumptions (p. 17) will generally be somewhat in excess of the true sum. Assuming that shop shaft and cylinder are on opposite sides of crank shaft, and that the belting makes an angle of 30° with the horizon, we can now combine the belt pull with the weight of the fly-wheel by the parallelogram of forces and note the angle made by the diagonal with the horizontal. The centrifugal force of the counterweight and the pressure of the connecting rod against crank-pin do not act in the same plane and should be given separate diagrams, if great exactness is desired. But such a refinement is here entirely unnecessary, and we may approximate by assuming them both to act in some convenient, intermediate, plane which is nearest to the larger of these two forces. These two may then be combined by the parallelogram of forces, again noting the angle the diagonal makes with the horizontal.

The bending forces have thus been reduced to two forces and to the reactions of the bearings. We now construct by the methods of graphical statics a diagram of bending moments for each of these forces.

Suppose that in Fig. 53 the scale of distances is $\frac{1}{4}$ th, or 3" to the foot, and the scale of forces is, say 200 lbs. to the inch. The polygon of forces due to the fly-wheel resultant found above, is ABC , the length AB representing the reaction of the out-board bearing and BC the reaction in the main bearing, the two reactions having the ratio $\overline{MN} + \overline{NQ}$. Choosing O in the polygon of forces on the horizontal through B , the closing line AD of the equilibrium polygon $ADGF$ will also be horizontal, and thus

convenient for subsequent combinations. The distance OB or the pole O from AC is chosen so as to give a convenient factor for multiplying the vertical chords or intercepts HI of diagram of moments $ADGF$. We know from the principles of graphical statics that the product of the intercept HI by the constant OB measures that part of the bending moment at cross section KL of shaft which is due to the forces on the fly-wheel. As OB is constant, the intercepts themselves may be taken to represent the bending moments of the cross sections of the shaft directly above them. The numerical value of any bending moment is found by multiplying the proper intercept by the distance $OB \times 200 \times \frac{1}{4}$, the last two factors respectively representing the scale of forces and the scale of distances on drawing.

If we take OB equal 4 inches, the product of these three factors $= 200$, which is a convenient factor with which to multiply directly the intercepts HI obtained from $ADGF$, the diagram of moments.

In like manner for the second resultant, due to counterweight and crank-pin pressures, we find the polygon of forces $A'B'C'O'$ and the equilibrium polygon $D'E'C'$.

This second resultant is constantly varying in direction and intensity with the rotation of the crank. It is greatest at the beginning of the forward stroke. But this does not correspond with the piston position at which the maximum twisting moment exists. As the greatest stress on the shaft probably occurs near this position, we shall suppose the diagram of moments due to this second resultant to be found for the crank position corresponding to the maximum twisting moment. The vertical intercepts of polygon $D'E'C'$ represent the bending moments, which are due to the action of this second resultant and its numerical value for any cross-section is found by multiplying by 200 the intercept of this diagram immediately below the cross-section in question.

These two diagrams of bending moments are both placed, for convenient combination, in the plane of the paper, though the forces calling them forth are neither parallel nor do they act in

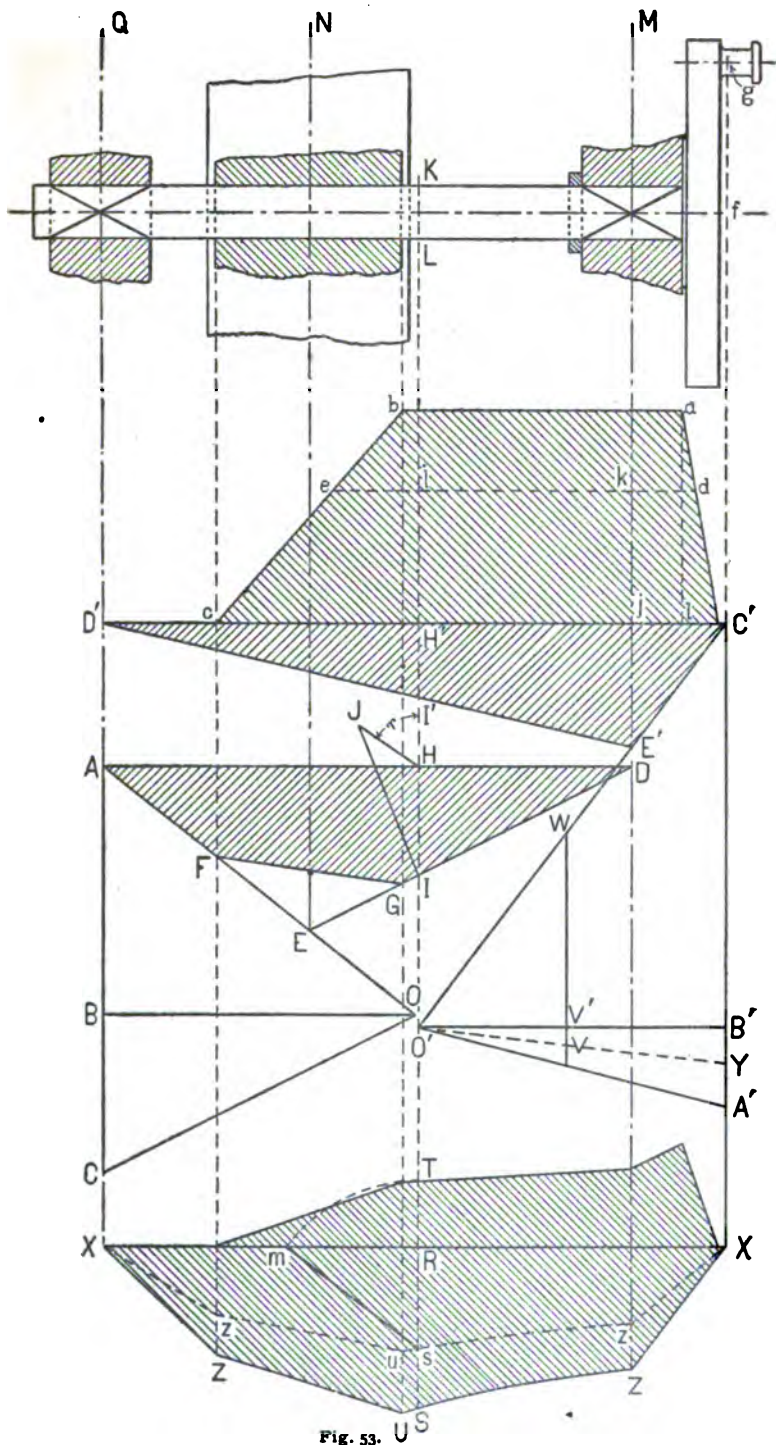


Fig. 53.

the same plane. As the intensities of the bending moments are represented by linear quantities and their directions are those of the forces producing them, they may be combined like forces, by means of the parallelogram of forces. This we will now proceed to do, being careful to take into account the angle between the forces. It will probably often happen that this angle is a small one, in which case the intercepts of the two diagrams may, with little error for the present purpose, be directly $\left\{ \begin{array}{l} \text{added} \\ \text{subtracted} \end{array} \right\}$ when the directions of the bending moments developed at any cross-section by the two resultants are $\left\{ \begin{array}{l} \text{alike} \\ \text{unlike} \end{array} \right\}$.

When the angle $\angle PHJ = T$, included between the arrow heads of the forces is large they should be combined on the principle of the triangle of forces. In the figure this is illustrated by laying off the length $H'P'$ of one diagram on HJ and joining J and I . We now take this line IJ of the second diagram and lay it off below the horizontal XX making it equal to RS , which is located on the vertical $SRIHH'$ passing through the assumed cross section KL . Repeating this process for every other cross section we get a third (the lowest) diagram $XXZZSZ$ whose intercepts represent the resultants of the purely *bending* moments attacking each cross-section.

The next step is to combine this resultant bending moment M_b with the twisting moment M_t , so as to get an equivalent or ideal bending moment M_{bi} . The formula connecting these three quantities is:

$$M_{bi} = \frac{3}{8} M_b + \frac{5}{8} \sqrt{M_b^2 + M_t^2}. \quad (73)$$

Before undertaking the graphical construction of this formula we will construct the diagram of twisting moments to the same scale as the bending moments. The twisting moment varies for the different crank-positions, but we will at once assume its maximum value $P_{\phi}R$. To have the intercepts of the diagram of twisting moments on the same scale we must make $P_{\phi} \times R = \overline{O'B'} \times y$. Making, in one of the polygon of forces,

$C'Y = P_{cf}$ and $O'V' = R = fg$, we get, by similar triangles, the distance $VW = \frac{P_{cf} \times R}{O'B'} = y$. Now lay off $la = y$ above the horizontal $C'D'$ and through a draw ab horizontal. The inclined line bec can be taken to represent the varying twisting moment of crank-shaft within the fly-wheel hub. The vertical intercepts of diagram $abcC'$ represent the twisting moment on the same scale as the other moments.

We can now return to the graphical construction of the last formula. The last term containing the radical can also be written

$$\sqrt{\left(\frac{5}{8} M_b\right)^2 + \left(\frac{5}{8} M_t\right)^2}.$$

The first term $\frac{5}{8} M_b$, under the radical is found by taking $\frac{5}{8}$ of each vertical intercept of the diagram $XXZUSZ$. For instance, $Rs = \frac{5}{8} RS$. The broken line in this diagram represents this division of $RS = M_b$ into $\frac{5}{8}$ ths and $\frac{3}{8}$ ths segments. In like manner $\frac{5}{8} M_t = kj = \frac{5}{8} la$ can be found. Making $Rm = \frac{5}{8} M_t = H'i$, the hypotenuse $ms (= sT)$ of the right-angled triangle mRs is the graphical equivalent of the last, radical, term of our formula. Adding it to $Ss = \frac{3}{8} M_b$ we get $ST = M_{bi}$. Its numerical equivalent, with the scales assumed above, is $M_{bi} = \text{length of } ST \times OB \times \frac{1}{4} \times 200 \text{ inch lbs.}$ This value of M_{bi} is to be substituted in the general formula for shafts subjected to bending stress

$$d = 2.17 \sqrt[3]{\frac{M_{bi}}{f_b}} \text{ inches,} \quad (74)$$

where f_b is the permissible working stress in lbs. per $\square''$ when shaft is subjected to bending stress. The result may be checked by the following empirical formula representing high-speed engine practice, for diameter of main, crank-shaft, journal

$$d = 0.44 D + \frac{1}{2}; \quad (75)$$

here D represents the diameter of the cylinder.

Having indicated the general method of procedure it will now be easy to apply the graphical solution to cases in which the fly-wheel overhangs the bed.

XI.

DETERMINATION OF PLANE OF DIVISION OF THE BRASSES AND LENGTH OF THE MAIN BEARING.

The plane of division of the brasses should be at right angles to the resultant of all those pressures which exist when the engine is running with its average load. These pressures are components of the weight of shaft and fly-wheel, pull of belt, centrifugal force of counterweight and the thrust or pull of connecting rod. For each crank position these four components may be combined by the polygon of forces so as to give the resultant force on bearing.

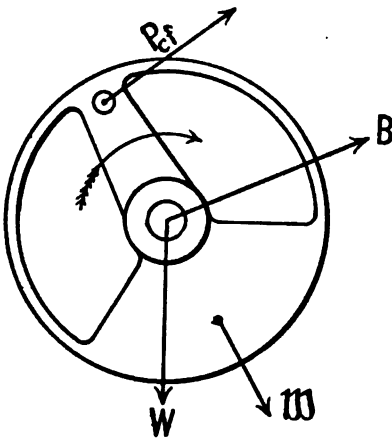


Fig. 54.

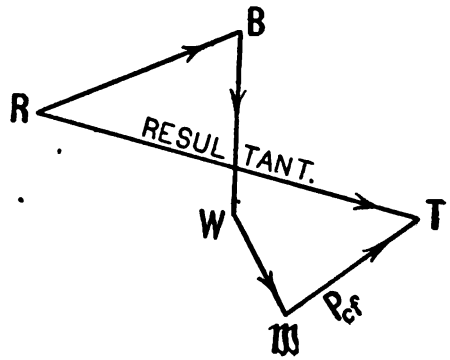


Fig. 55.

In Figs. 54, 55 and 56 these forces are shown and their force polygon. For a given load, speed and engine, RB and BW may be regarded as constant in direction and intensity, consequently, RW may have an invariable position on the diagram Fig. 56. As the action of the counter-weight is constant in intensity and opposite to the crank in direction, we may take W as the center of a circle described with Wm as a radius. From the points of this circumference we draw mT equal and parallel to the (reduced)

values of P_c corresponding to the different crank positions, then will RT represent the resultant of the two pressures exerted by the shaft against its main bearing. Fig. 56 shows two determinations, RT and RT' , of these resultant pressures against the bearings for the 30° and 60° crank positions.

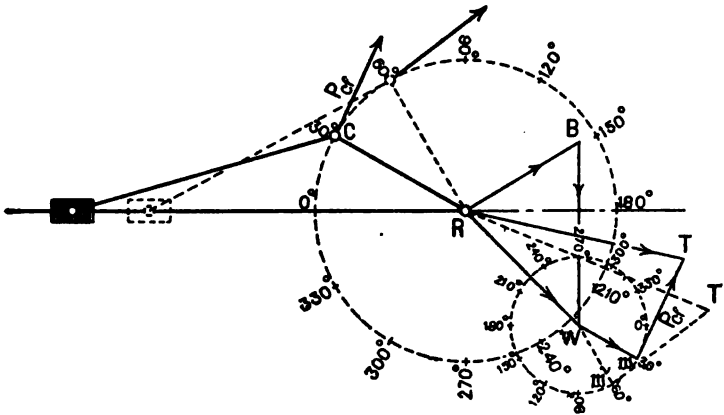


Fig. 56.

The forces in this figure (56) have been chosen at random and do not represent any special case. When a graphical determination, Fig. 53, of the diameter of the crank shaft has been made, the components of these pressures at the main bearing can be found from the force polygons there drawn. A resultant should be drawn for each of a series of equidistant crank positions and the plane of division of the brasses placed at right angles to the resultant of these resultants. Great accuracy is not necessary, in most cases inspection will determine best position of plane.

For the determination of the length l of crank-shaft journal of a high speed engine the following empirical formula, representing good practice, may be used

$$l = 1\frac{1}{2} d + 3'' . 0 \quad (76)$$

where d is the diameter of same journal determined according to another empirical formula, for same class of engines,

$$d = 0.44 D + \frac{1}{2}'' ,$$

D representing the diameter of piston.

XII.

DETERMINATION OF THE DIMENSIONS OF STEAM PASSAGES.

The first dimension to be settled in valve-gear problems is the requisite area of the ports, for upon this depends the maintenance of the desired steam pressure. The principal factor in its determination is the velocity of the steam current and as this varies with the piston's motion in different parts of the stroke, we must see to it that the minimum cross-section in the passage is adequate for every part of the stroke. This part of the problem divides itself into two parts: (*a*) the ascertainment of the minimum area needed at the given piston speed and with given cylinder area, and (*b*) an examination of the cross-sections of the steam passages that are really available or effective with the special type used, taking into account the contraction caused by the flow past edges and the narrowing of ports caused by the motions of the valves on each other or on their valve seats.

This second part presents no theoretical difficulties and we shall therefore not examine any special case here, but this should be done in the draughting room when the type of valve has been chosen. It is practically important in case there are supplementary, or multiple, ports in the main or distribution valve. Allowance should also be made for the contraction of the steam current when the ports are very narrow, for this effects a very notable reduction of the port area geometrically available. We will now consider the minimum port area necessary for a given area of cylinder and a given piston speed.

Radinger's criterion of sufficient passage area was whether or not the admission line of indicator card remained horizontal up to cut-off.

According to Radinger's experiments,

$$\frac{\text{Area of port}}{\text{Area of cylinder}} = \frac{bl}{A} = \frac{\text{average speed of piston in feet per sec.}}{100} \quad (77)$$

This rule evidently corresponds to an average velocity of 100 feet per second of the current of steam in the port; but the experiments seem to have been made on engines in which the maximum cut-off was equal or greater than 0.5, consequently if we assume that in the engines experimented upon the ratio of connecting rod to crank varied from 4 to 6 we can easily determine the maximum permissible velocity of the steam in the ports.

For when $\frac{\text{connecting rod}}{\text{crank}} = 4$ to 6 we have the ratio of

$$\frac{\text{Max. velocity of steam in ports}}{\text{Ave. velocity of steam in ports}} = 1.62 \text{ to } 1.59 \text{ respectively.} \quad (78)$$

This gives about 160 feet per second as the maximum permissible velocity of the current of steam in the ports when the steam passages are long and narrow. Radinger says that when the steam passages are short and the cut-off short (for instance, when there are separate admission valves for each end of the cylinder) somewhat smaller port areas may be employed than would result from above rule; in other words, in such cases the maximum permissible velocity of the current of steam may be somewhat greater than 160 feet per second.*

Mr. Charles T. Porter gives the rule that the velocity of the current of steam in the ports should not exceed 200 feet per

* Some American engineers prefer the rule that the *average* speed in short steam ports should not exceed 150 feet per second, and in the exhaust ports should ≤ 125 feet per second. When the steam is admitted and discharged through the same passage, some engineers recommend that its cross-section be proportioned for the exhaust, preferring an increase of clearance to an increase of back pressure.

TABLE XIII.

VELOCITY OF PISTON FOR A UNIT OF VELOCITY OF CRANK-PIN.

To obtain actual velocity of piston multiply tabular quantity by actual velocity of crank-pin.

Forward stroke is towards, and return stroke, away from, crank shaft.

Crank Angles.		Connecting Rod ÷ Crank =					
Forwa'd.	Return.	4.0	4.5	5.0	5.5	6.0	∞
5	175	.1089	.1064	.1045	.1030	.1016	.0832
10	170	.2164	.2117	.2079	.2047	.2022	.1737
15	165	.3215	.3145	.3089	.3054	.3005	.2588
20	160	.4227	.4136	.4065	.4019	.3957	.3420
25	155	.5189	.5081	.4995	.4925	.4866	.4226
30	150	.6091	.5968	.5870	.5791	.5724	.5000
35	145	.6923	.6788	.6682	.6596	.6523	.5736
40	140	.7675	.7533	.7421	.7329	.7253	.6428
45	135	.8341	.8195	.8081	.7988	.7910	.7071
50	130	.8914	.8771	.8657	.8564	.8488	.7660
55	125	.9392	.9253	.9144	.9055	.8982	.8192
60	120	.9769	.9641	.9540	.9458	.9390	.8660
65	115	1.0046	.9932	.9842	.9769	.9709	.9063
70	110	1.0224	1.0127	1.0052	.9990	.9939	.9397
75	105	1.0304	1.0228	1.0169	1.0121	1.0082	.9659
80	100	1.0289	1.0237	1.0199	1.0164	1.0137	.9848
85	95	1.0186	1.0160	1.0139	1.0127	1.0109	.9962
90	90	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
95	85	.9738	.9764	.9785	.9797	.9816	.9962
100	80	.9407	.9460	.9500	.9532	.9559	.9818
105	75	.9016	.9091	.9150	.9198	.9237	.9659
110	70	.8571	.8667	.8743	.8804	.8855	.9397
115	65	.8080	.8194	.8285	.8357	.8418	.9063
120	60	.7552	.7680	.7781	.7863	.7931	.8660
125	55	.6992	.7130	.7239	.7328	.7401	.8192
130	50	.6407	.6550	.6664	.6756	.6833	.7660
135	45	.5801	.5958	.6061	.6155	.6232	.7071
140	40	.5181	.5323	.5435	.5527	.5603	.6428
145	35	.4549	.4683	.4790	.4876	.4949	.5736
150	30	.3909	.4032	.4130	.4209	.4276	.5000
155	25	.3264	.3371	.3458	.3528	.3586	.4226
160	20	.2614	.2704	.2776	.2821	.2884	.3420
165	15	.1962	.2032	.2088	.2123	.2171	.2588
170	10	.1309	.1356	.1394	.1426	.1451	.1737
175	5	.0655	.0679	.0698	.0714	.0727	.0832

second. This is what we shall assume as suitable for short passages. Representing the actual velocity of the steam by V_s , and by b' the minimum permissible opening of port when the crank makes an angle ω with the line of centers, we have the following formula when the steam passages are long and narrow:

$$\frac{b' \times l}{A} = \frac{V_s}{160} \text{ or } b' = \frac{A}{l} \frac{V_s}{160} \quad (79)$$

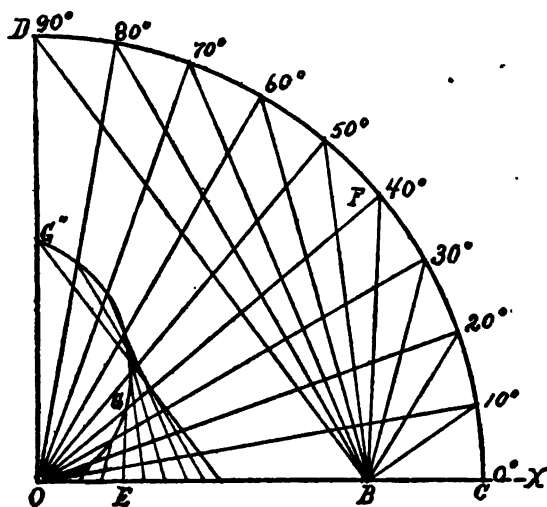
$$\text{or } b' : \frac{A}{l} = V_s : 160$$

when the passages are short, we have

$$\frac{b' \times l}{A} = \frac{V_s}{200} \text{ or } b' = \frac{A}{l} \frac{V_s}{200} \quad (80)$$

$$\text{or } b' : \frac{A}{l} = V_s : 200 \quad (81)$$

The opening of the port b' corresponding to the different crank angles may now be graphically determined (Fig. 57) as follows: Assuming that the area of cylinder A and length of port l are



In this diagram $\overline{OC} = \frac{A}{l} = \frac{\text{area of cylinder}}{\text{length of port}}$
Fig. 57.

given, and finding the velocity of the piston from Table XIII on page 128, we first draw a series of radii, making the angles 10° , 20° , 30° , etc., with the line OX . These angles correspond to the crank angles. We now describe a circle CD with O as a center, and $\frac{A}{l} = OC$ as a radius. We must lay off OB on OX and equal either to 160 or 200, according as the steam passages are long or short. To find the minimum port opening b' , corresponding to a particular position OF of crank, we join the point B with F (the intersection of the circle CD with the radial line OF , representing the crank angle in question) and then lay off OE on OX , equal to the corresponding velocity V_s . We now draw through E a line EG parallel to BF , then because $OF = \frac{A}{l}$, $OE = V_s$, $OB = 160$ or 200 .

$$OG : OF = OE : OB.$$

$$OG : \frac{A}{l} = V_s : 160 \text{ or } 200, \quad (82)$$

comparing this proportion with the preceding ones we see that $OG = b' =$ minimum opening of port for crank-position OF ; in like manner all values of b' can be determined and a curve OGG'' may be drawn through the extremities of OG'' , OG , etc., which will represent the values of b' ; the portion of the radius-vector included between the curve OGG'' and the pole O will give the value of b' for the corresponding crank angle.

When b' and $\frac{A}{l}^*$ are drawn full size, V_s and 160 should each be laid off to same scale, say 20 feet (per second) to the inch. If $\frac{A}{l}$ be drawn on an $\frac{1}{s}$ scale while b' is drawn full size we must lay off 160 on a scale of $s \times 20$ feet to the inch while V_s is laid off

* $\frac{A}{l}$, it should be noted, is a linear dimension and is expressed in inches.

as before, to a scale of 20 feet to the inch. When main and expansion valves are of the ordinary type the value of b' obtained by the above diagram may be laid off directly upon the main and expansion valve diagrams and compared with the actual openings of port given by the latter. But when the valves are of the gridiron type having two admission ports instead of one, we must halve the values of b' obtained from the above diagram and thus reduced lay them off on the valve diagram for comparison. If the expansion valve is of the piston type we must lay off the values of b' on a scale equal to $\frac{l}{\pi d'_1}$ (d'_1 = diameter of piston valve), for the minimum opening b'' of piston-valve port (*i. e.* the minimum for a particular crank angle) must equal

$$b'' = \frac{l}{\pi d'_1} b' \quad (83)$$

The *full* width b''' of port for piston valve may of course be found by substituting for b' its maximum value b in the preceding formula. Formula (83) assumes that the steam passage within the main valve has everywhere a cross section $\geq$ than the maximum, annular, port opening effected by the piston valve.

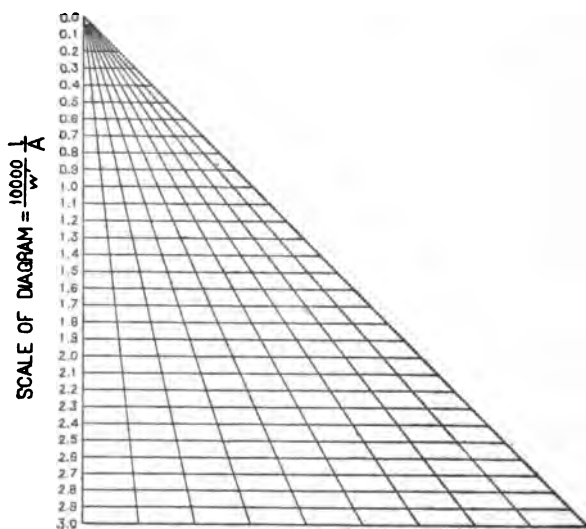
As regards the steam passage itself (not the port) if it is large enough for the exhaust it will be large enough for the admission. Its area should therefore be the quotient,

$$A \times \frac{\text{Average piston speed (ft. per sec.)}}{100 \text{ to } 125} \quad (84)$$

The port opening for exhaust should also be tested, taking 200 feet per second as the maximum velocity of the current that is permissible.

By plotting the port-openings as ordinates on the piston stroke as a base, and connecting them by a curve, the slope of this curve where it crosses the base will be a measure of the rapidity of cut off.

Where many valve-gear problems must be solved, as in a college draughting-room, two diagrams like Fig. 57 may be drawn which will cover the whole range of practice. In both these general diagrams the ratio of connecting rod to crank = 6, but the maximum permissible velocity of steam is 200 feet per second in one diagram and 160 feet per second in the other, corresponding to short and long steam passages respectively. For these general diagrams assume $\frac{A}{l} = 10$ and average speed of piston in feet per minute (w') equal to 1000. The diagrams will then be applicable to any problem, giving the minimum port-openings $OG (= b')$ to a scale of $1 \div \frac{Aw'}{1000l}$ that is, $\overline{OG}$ on the diagram will be $\frac{10000}{w'} \frac{l}{A}$ the true size. In this way a diagram whose scale varies with the assumed data can be made to fit all cases. To reduce the port-openings given by these diagrams to their full size values on the Zeuner, valve-circle, diagram a reduction



PORT OPENING GIVEN BY DIAGRAM.

Fig. 58.

arrangement, like that shown in Fig. 58, will be found very convenient. The full sizes are taken from horizontal line 1.0.

As regards the thickness of the cylinder walls, Prof. W. C. Unwin* says, "(a) that it should be strong enough to resist the internal steam pressure; (b) rigid enough to prevent any sensible alteration of form; (c) it must be thick enough to insure a sound casting; (d) thick enough to permit rebor-ing once or twice when worn. Generally other considerations than strength are of so much importance, that the empirical rule agrees better with practice than a rule making the thickness t depend upon steam pressure:" hence when D is diameter of cylinder

$$t = 0.02 D + 0.5 \text{ to } 0.02 D + 0.75. \quad (85)$$

XIII.

VALVE DIAGRAMS AND DIMENSIONS OF VALVE AND GEAR.

We shall assume that the reader is acquainted with the principal functions and common varieties of valves, and also with the terms designating the most important dimensions and parts of ordinary valve gear. This is usually given in elementary text-books on the steam engine. There is such variety in valves in the matter of arrangement of steam passages, in the method of balancing and in the division or assignment of functions, and all this involves so much special dimensioning that it would need a separate treatise to do justice to this part of the subject. We can here only attempt to point out the principal types and the methods of ascertaining the leading dimensions.

The ultimate object of all valve-gear discussion is to establish the relation existing between movement of piston and movement of valve. The piston's motion has already been fully discussed on pp. 35-39, and a table on pp. 26-27 gives its travel for different crank positions. After the travel of the valve has been ascertained for these same crank positions it will be easy to construct a diagram giving directly the desired relation.

* Elements of Machine Design. Eleventh Edition, Part II, pp. 33.

The valve motion that has been most extensively used in practice is that which is known to mathematicians as "harmonic motion," and to engineers as, the motion of the slotted cross-head. Almost all valve-motion deviates a little from this, owing to the angular motion of the eccentric rod. Of late years a few valve-gears have been devised in which the valve's movement differs considerably from that corresponding to the "harmonic law;" but as yet these gears have not come into extensive use. They have usually complicated mechanisms that do not follow any simple law in their valve-movements and are therefore not readily amenable to scientific treatment. We shall confine ourselves almost wholly to the first, widely used, class. Link-motions of the common forms do however come within this class, but only the Porter-Allen (Fink) form is much used in stationary, high-speed, engine work. In this section only this latter form of link motion will receive a discussion. The Appendix will contain a general, graphical, method for obtaining the valve diagrams of the common forms of link-motions.

The valve-gears subject to the "harmonic law" can be divided into two groups:

- a. Gears with but a single valve which slides on a stationary seat.
- b. Gears with two valves, of which one slides on the other.

The first of these will be considered under the two heads of invariable steam distribution and variable steam distribution.

In second group the main part of our problem will be to show that the complex mechanism producing the desired relative motion of the valve on its seat, is equivalent to the action of one, single, eccentric capable of giving the valve in question an identical, absolute, motion, on a similar, but stationary, seat. The problem before us is therefore mainly a kinematic one. In order that we may simplify the usually difficult and complicated parts of this subject it will be necessary to review the elementary parts and present them in a new light.

SINGLE-VALVE GEAR.—INVARIABLE STEAM DISTRIBUTION.

We have already stated that the main object of valve gear discussion is to find the relation between piston and valve movement, and that the piston's position for various crank angles has already been ascertained by suitable formulas and tables. As the eccentric is nothing but a short crank, a similar relation, of course, exists between the valve-slide positions and eccentric angles, but with this difference, that in the former case the ratio of crank to connecting rod is a comparatively large fraction, say $\frac{1}{6}$, while in the latter case the ratio of eccentric radius to eccentric rod is a very small fraction, often less than $\frac{1}{60}$. In the former case the piston's motion is unsymmetrical in the first and second halves of its stroke, while the valve's motion is practically symmetrical on each side of its middle position and for an infinitely long eccentric rod would be perfectly so, and this is the assumption which is always made in this sort of discussions and which will therefore underlie all our subsequent work. Moreover, it is practically most convenient to estimate piston travel from the beginning of its stroke, while valve travel is best estimated from its center of motion or mid-position, because of the symmetry of its motion and of the arrangement of the ports. For these two reasons the expressions for piston and valve travel are usually different. Just at present we will confine ourselves to the construction of a diagram representing the distance of the valve or slide from its mid-position at the different crank angles and leave the graphical representation of the piston's travel to the moment when we shall combine the two movements into one diagram and thus exhibit graphically their desired relation.

We shall first treat the slide-valve as a simple, rectangular, plate, moving on an unperforated seat and driven by an infinitely long eccentric rod. In Fig. 59, let O be the center of the engine shaft, OC_1, OC_2, OC_3, OC_4 different crank positions and OE_1, OE_2, OE_3, OE_4 , the corresponding eccentric positions, the eccentric

being set ahead of the crank by a constant angle C_1OE_1 . Let NZ be the dead center line of the eccentric OE (which may or may not coincide with that of the crank OC), then will OP be the mid-position of the eccentric. When the eccentric is in this position the valve will also be in its mid-position. When the eccentric is in position OE_1 , the slide will be at the distance E_1S_1 from its mid-position and this distance we will call the travel

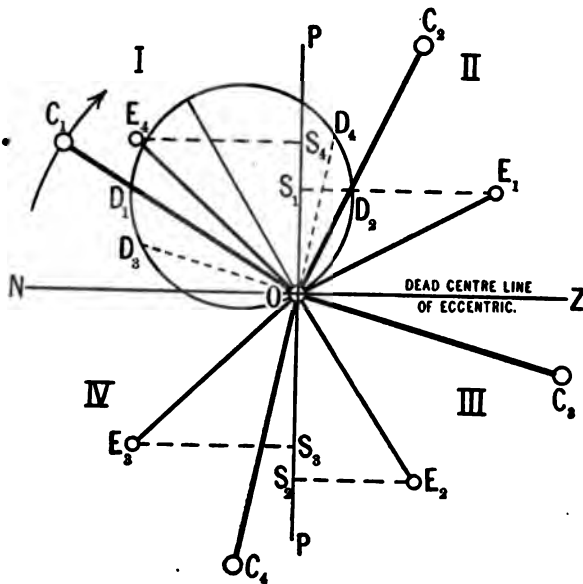


Fig. 59.

of the slide. For the other eccentric positions OE_2 , OE_3 , OE_4 , the slide-travel is E_2S_2 , E_3S_3 , E_4S_4 , respectively. If now we lay off this travel ES on the corresponding crank positions while the eccentric is to the right of OP and on the prolongations of the corresponding crank positions when the eccentric is to the left of OP , and making $OD_1 = E_1S_1$, $OD_2 = E_2S_2$, $OD_3 = E_3S_3$ and $OD_4 = E_4S_4$, we get the curve $OD_1D_2D_3D_4$, which is called the polar diagram of the slide motion. It is evident from the method of construction that the chord cut by this curve from the

crank or its prolongation is the travel of the slide from its middle position. It is also evident that the slide is respectively to the $\left\{ \begin{smallmatrix} \text{right} \\ \text{left} \end{smallmatrix} \right\}$ of its middle position when the curve cuts the $\left\{ \begin{smallmatrix} \text{actual} \\ \text{prolonged} \end{smallmatrix} \right\}$ crank, and that the slide is going $\left\{ \begin{smallmatrix} \text{away from} \\ \text{towards} \end{smallmatrix} \right\}$ its middle position when these chords $\left\{ \begin{smallmatrix} \text{increase} \\ \text{decrease} \end{smallmatrix} \right\}$.

With an infinitely long eccentric rod, the curve $OD_3D_1D_4D_2$ becomes a circle. To prove this let us take a general case, Fig. 60, in which dead centre line OX of crank makes an angle XOZ with the dead centre line OZ of the eccentric. From the right hand half of NOZ , and in a direction *opposite* to the crank's rotation, measure the angle $ZOE = \sigma$ between this dead center OZ and the eccentric position OE that corresponds to any crank position OC . Then lay off from the crank OC in the *same* direction as the rotation, this angle $\sigma = COD$. Make $OD = OE$ and on OD as a diameter describe the circle $OF'D$. The right-angled triangles OEF and ODF' are evidently equal, hence $OF' = OF = ES =$ valve travel, that is, the circle $OF'D$ cuts a chord OF' from any crank position OC that is equal to the

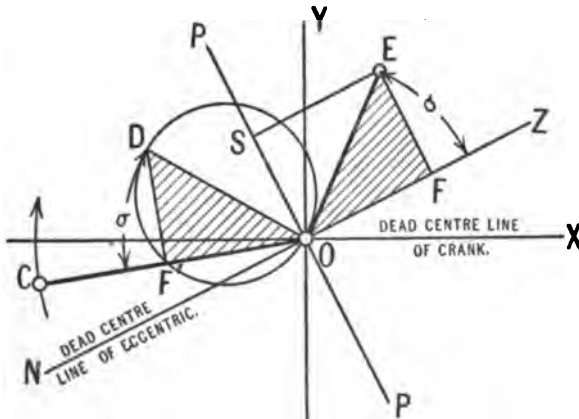


Fig. 60.

valve travel ES for that instant. As the polar co-ordinates of circle $OF'D$ and of the polar curve $OD_3D_1D_4D_2$ (Fig. 59) are alike, the latter is also a circle, and may be called the slide- or valve-circle.

By reversing the steps of this construction we may evidently find the position of the eccentric from the crank position, when the valve circle is given. Our rule then is, to measure from the crank, in the direction of its rotation, the angle σ included between it and the diameter of the valve-circle, and then lay off this angle σ from the dead center line OZ , in a direction *opposite* to the rotation of the crank, and this will give the desired position of the eccentric.

The angle $ZOE = \sigma$ is estimated from the portion of the line OZ to the *right* of center O , and so long as the eccentric (or slide) is to the *right* of its middle position the angle made by the eccentric with the portion OZ of the reference line will be less than 90° . When this is the case the diameter OD of the slide-circle is always less than 90° from the crank position, which ensures that this circle will cut the $\left\{ \begin{array}{c} \text{actual} \\ \text{prolonged} \end{array} \right\}$ crank whenever the eccentric or slide is to the $\left\{ \begin{array}{c} \text{right} \\ \text{left} \end{array} \right\}$ of its middle position. This is true for all values of σ , for all positions of the crank, for either right- or left-handed rotation, and whether slide-seat is to the right or left of shaft O , provided, of course, that there is no reversing lever between eccentric and slide.

The position of this valve-circle does not change while the crank rotates, its diameter making with the fixed reference line OZ an angle DOZ that is equal to the constant angle COE between crank and eccentric. That this is true is evident from:

$$COE = COD + DOE = ZOE + DOE = ZOD.$$

The valve-circle's position is therefore dependent only on the relative position of eccentric to crank. Provided this eccentric setting remains exactly the same in direction and magnitude, the crank may have either right-handed or left-handed rotation,

may occupy any position in any one of its quadrants and yet give the self-same valve circle whenever the construction described above is strictly followed.

For a given eccentric setting the valve-circle will have the same position, that is, the slide's distance, position and direction of motion relatively to its center of travel will always be the same for the same crank position. But when the eccentric setting is different, the motion of the slide relatively to the center of travel will be different, even though the crank position and rotation are exactly the same. This is illustrated in Fig. 61:

- I. Slide is to the right of its middle and is going towards it.
- II. Slide is to the left of its middle and is going towards it.
- III. Slide is to the left of its middle and is going away from it.
- IV. Slide is to the right of its middle and is going away from it.

The corresponding valve circles are given in the second row of Fig. 61 when the crank position is OC and the rotation right-handed, and in the third row when crank position is OC' and the rotation left-handed. The angle between the crank and eccentric being the same in magnitude for both rotations, the laying off angle σ , used in finding location of diameter of valve circle, will be the same for the corresponding figures of the second and third row.

In the last row of Fig. 61 we have the same four, possible, types of slide valves which are kinematically exactly like the corresponding Cases I, II, III, IV of the first row, and have the same valve circles. The eccentric driving these slide valves is not shown because convenience of illustration requires the valve to be placed above the cylinder and because the setting of the eccentric relatively to crank is shown in the first row for both right- and left-handed rotation. Each valve is closing left port.

Each of these four types has its special, valve, characteristics. A valve is said to have $\left\{ \begin{array}{c} \text{positive} \\ \text{negative} \end{array} \right\}$ lap when the port is $\left\{ \begin{array}{c} \text{closed} \\ \text{open} \end{array} \right\}$ in the valve's mid-position. We will call a valve

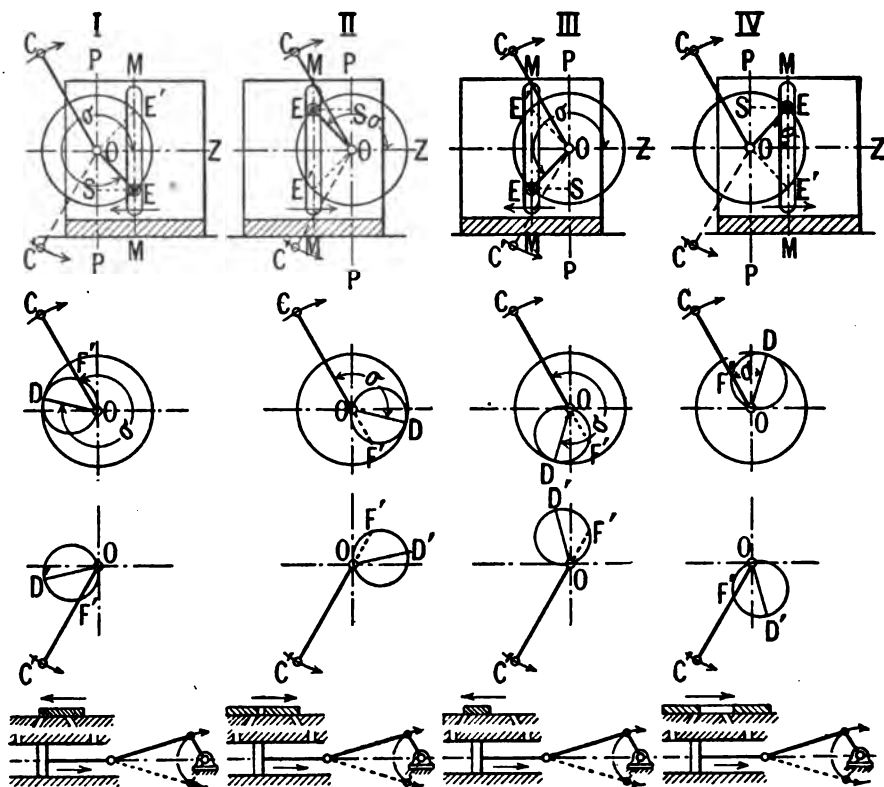


Fig. 61.

edge $\begin{cases} \text{direct} \\ \text{indirect} \end{cases}$ when it closes the *left* port (or passage leading to left end of cylinder) by moving to the $\begin{cases} \text{left} \\ \text{right} \end{cases}$. When *all* edges of the valve are $\begin{cases} \text{direct} \\ \text{indirect} \end{cases}$ we will call the valve itself $\begin{cases} \text{direct} \\ \text{indirect} \end{cases}$. When some of the edges are direct and others indirect (as in the ordinary **D** valve where the steam edges are direct and the exhaust edges indirect), we will call the valve itself direct or indirect according as the edges regulating *admission* of steam are direct or indirect.* These definitions hold whether the ports are

* English and American engineers, so far as the writer is aware, have never used these terms, *direct* and *indirect*. WIEBE makes use of them in his "Darstellung der Verhältnisse der Schieberbewegung." They are short, clear, terms and may profitably be employed in this extensive subject.

in the seat or the face of the valve. In Fig. 61, fourth row, we have

- I. A valve with positive lap and direct cut-off.
- II. A valve with positive lap and indirect cut-off.
- III. A valve with negative lap and direct cut-off.
- IV. A valve with negative lap and indirect cut-off.

We will designate them briefly as, I positive direct, II positive indirect, III negative direct and IV negative indirect, valves.

The ordinary **D** valve in such extensive use is positive direct on its steam side and positive indirect on its exhaust side. The Meyer expansion valve with its halves placed so close together that in the middle position they fall within the outer edges of their own steam ports is an example of a negative direct valve, and the same expansion valve when its cut-off plates are placed so far apart that in the middle position of the valve its inner edges are outside of the inner edges of their own steam ports, is an example of the negative indirect valve.

Other positions of these halves between these extremes will give positive direct and indirect types. With the same eccentric setting therefore it is possible for a slide to represent any one of the four types of valves, according to the position of the pair of ports relatively to the middle position of the slide. (But with the same eccentric setting and same throw of valve the cut-off will be different for each type.) The terms positive and negative, direct and indirect, refer only to this relative position of ports to valve and are entirely *independent of the eccentric setting*. (When the valve controls only one port, instead of two, the terms direct and indirect disappear entirely, only the distinctions positive and negative then remaining.) The angle between crank and eccentric, measured from the former in the direction of rotation, for any one of these valve types, may vary from 0° to 360° , although commonly each type has its eccentric set in the eccentric-setting-quadrant shown in Fig. 61.

In all of the engines of the fourth row of Fig. 61, for the same rotation, the same cut-off is taking place during the same stroke.

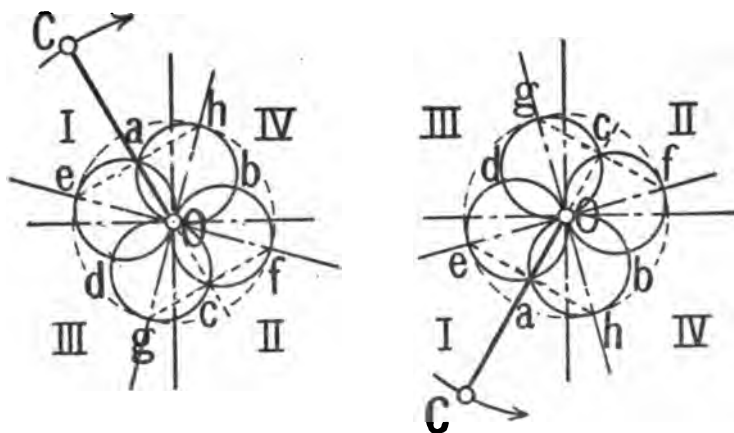


Fig. 62.

Hence four different types of valves can be arranged to accomplish the self-same object. The eccentrics which drive them will have four corresponding settings relatively to the crank. There will be four different valve-circles to represent the motion. They are grouped together in the two diagrams of Fig. 62, which differ only in the direction of crank rotation.

To find the valve-circle belonging to any type we draw the crank position OC corresponding to the cut-off effected by the valve, and prolong it beyond center O . Then lay off the lap, $Oa = Oc$, and draw circle $chfg$ with the radius of eccentric. At a and c erect the perpendiculars hae and gcf and thus determine the ends e, h, f and g of the diameters Oe, Of, Og , and Oh of the four valve-circles. To find the one belonging to the type of valve under consideration, notice whether this valve, when at cut-off, is to the right or left of its middle position on its seat. If it is to the $\left\{ \begin{smallmatrix} \text{right} \\ \text{left} \end{smallmatrix} \right\}$ one of the two circles $\left\{ \begin{smallmatrix} \text{I, IV} \\ \text{II, III} \end{smallmatrix} \right\}$ will be the desired one, because these cut the $\left\{ \begin{smallmatrix} \text{actual} \\ \text{prolonged} \end{smallmatrix} \right\}$ crank. This now reduces the choice to one of two circles. Now notice whether the valve in its cut-off position is moving towards or away from its middle position. If it is moving $\left\{ \begin{smallmatrix} \text{towards} \\ \text{away from} \end{smallmatrix} \right\}$ this mid-position the chord Oa (or Oc) of the proper valve-circle will $\left\{ \begin{smallmatrix} \text{diminish} \\ \text{increase} \end{smallmatrix} \right\}$ as the crank continues the rotation from its cut-off position. In Fig. 62 chord

Ob is accidentally equal to chord or lap Oa , but it is not so generally.

Hitherto in valve-gear discussions two diametrically opposite valve-circles were employed for they helped to show the main occurrences in the steam distribution at both ends of the cylinder. But in designing valve-gears we usually pass from the valve-circle to the actual gear and there is then danger of confusion as to the eccentric setting when two circles are used. There can be no objection to the use of two circles provided they can be distinguished by some means, say, by drawing the representative one in full lines. We shall however make use of but one valve-circle when there is but one crank. When there are two cranks, diametrically opposite, as in the Westinghouse engine, we shall use two valve-circles.

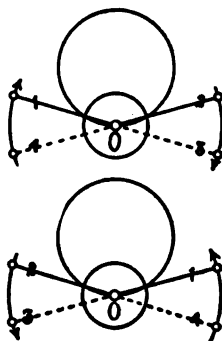
We will not here describe the occurrences of the steam distribution, as this is given in many of the elementary treatises on the steam engine and we have supposed the reader to be familiar with the elements of this subject. Besides, the occurrences for the ordinary **D** valve are inscribed on the Zeuner valve diagram in Fig. 63. But the following table may be of assistance when some of the unfamiliar valve-types are under consideration.

TABLE XIV.

PERIODS DURING WHICH PORT CONTROLLED BY VALVE IS OPEN OR CLOSED, (GIVEN BY THE LIMITING AND INTERMEDIATE CRANK POSITIONS.)

Condition of Port.		Type of Valve.*			
		I Positive Direct.	II Positive Indirect.	III Negative Direct.	IV Negative Indirect.
Right Port	Open	3-4	1-2	2-3-4-1	4-1-2-3
	Closed	4-1-2-3	2-3-4-1	1-2	3-4
Left Port	Open	1-2	3-4	4-1-2-3	2-3-4-1
	Closed	2-3-4-1	4-1-2-3	3-4	1-2

* Left port closed by positive direct valve when crank is at 2.
 " " " " indirect " " " " 4.
 " " " " negative direct " " " " 3.
 " " " " indirect " " " " 1.



This table is applicable to both exhaust and admission by considering the lap circles of the adjacent figures to represent in the former case the inside, exhaust, lap, and in the latter case the steam lap, provided account is taken of the valve type of the edge in question. Intercepts of lap and valve-circle measure openings.

As the valve circle gives us the valve travel to the right or left for any crank position and Tables, pp. 26-27, the corresponding piston-position, we can represent graphically by an oval sort of curve the desired relation between piston movement and valve-travel. The ordinates of this curve are the chords cut from the crank by the valve-circle and the abscissas are the distances of piston from end of stroke given by Tables V and VI. Lines drawn parallel to the base line, and above and below the latter a distance equal to the outside and inside laps, give the principal occurrences in the steam distribution. The diagram is easily drawn when the valve circle has been found.

There is another diagram which requires still less work and gives readily the actual piston and valve travel for any crank position. Let $OA = OB$, Fig. 63, equal the crank radius to some reduced scale. For convenience we will take the length of the connecting-rod three times that of crank. Then with the length $\overline{CB} = 3 \overline{OB}$ of this rod describe the arc EBF and with the same radius strike off through A an equal and parallel arc GAH . The horizontal lines included between these arcs will of course be equal to the stroke, and the horizontal intercepts between these arcs EF and GH and the crank-pin circle $AJLBU$ will represent exactly the distances of the piston from one or the other end of the stroke. For instance, when the crank-pin is at J , the intercept IJ will represent the distance s of the piston from the left end of the stroke. This is evident from equation (20) p. 35, where

$$s = R(1 - \cos \omega) + L(1 - \cos \alpha)$$

Here the first term of the second member is equal to $AZ = WJ$ and the second term of this member is equal to $VA = IW$.

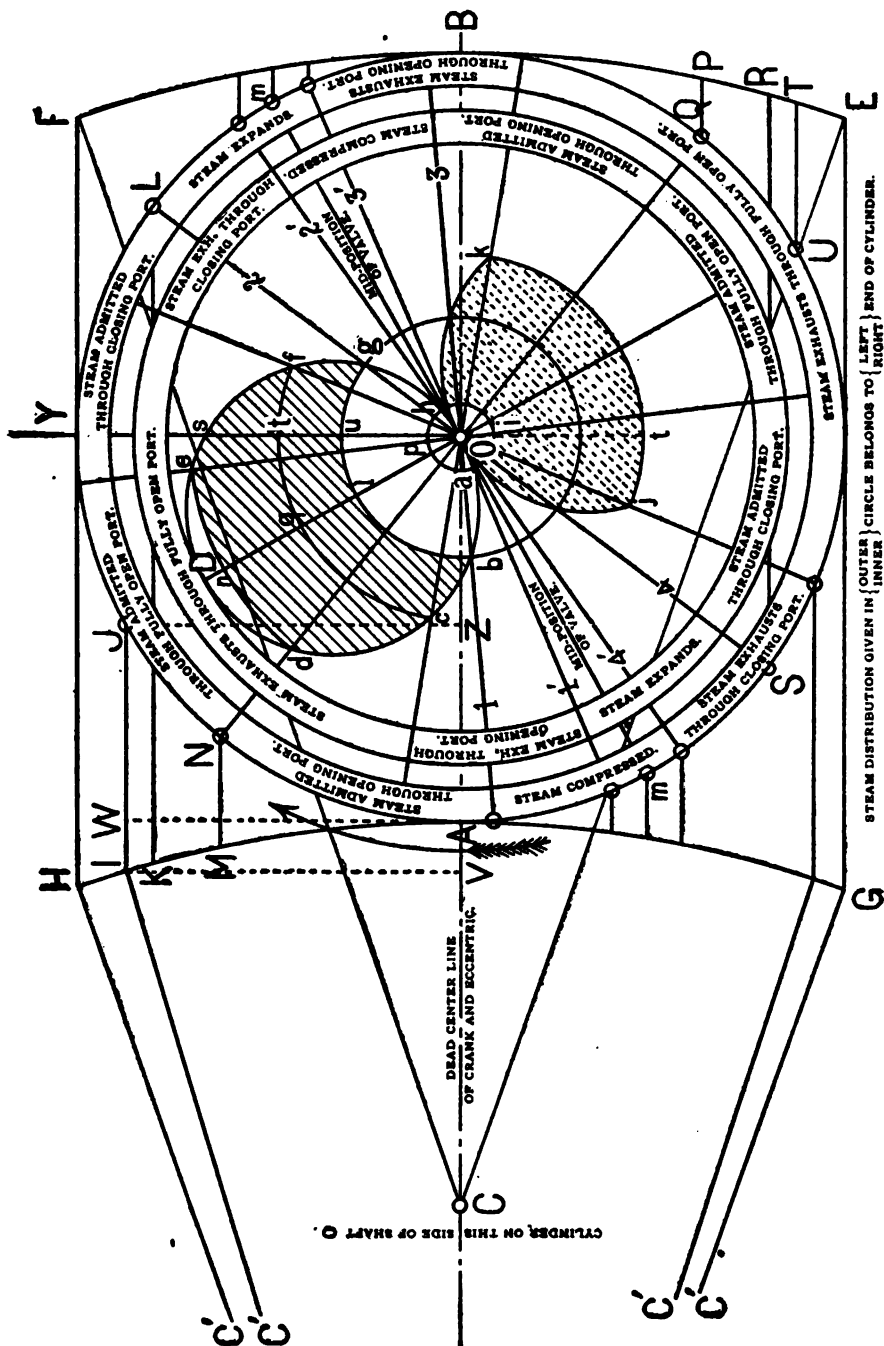


Fig.63.

When the crank is at the cut-off positions OL and OS , the piston travel is KL and RS for forward and return strokes respectively, and will be found to be unequal. The valve travels for the same instants are equal to Og ; the scale for valve travel will usually be different from that for piston travel. At another crank position, ON , the piston travel is given by MN and the valve travel by Od .

When the crank or its prolongation traverses the $\left\{ \begin{array}{c} \text{sectioned} \\ \text{dotted} \end{array} \right\}$ area, one or the other of the steam ports is open for $\left\{ \begin{array}{c} \text{admission} \\ \text{exhaust} \end{array} \right\}$.

When crank is at OY , the distance of the piston from beginning of the stroke is HY , the valve travel is Og , the opening of the left steam port is us and is closing, while at the other end of the cylinder the steam is exhausting through a fully open port, pt . In Fig. 63 the four crank positions 1, 2, 3, 4, are drawn which correspond to the opening or closing of the steam ports by the admission edges of the valve; the crank position $1^1, 2^1, 3^1, 4^1$, correspond to the opening or closing of the steam ports by the exhaust edges of valve. This diagram is for a positive valve. It will be a profitable exercise to closely compare the steam distribution given by Table XIV with that given by Fig. 63.

An ordinary **D** valve is usually set by giving it equal leads at the dead points of the crank; but this makes every other occurrence of the steam distribution take place unequally for the two strokes. We have already pointed out that there is no necessity for equality of lead; that the lead may be unequal provided the minimum value is sufficient to maintain the steam pressure, and cushion properly; nor is a positive lead necessary, there are cases when a negative lead is justifiable and desirable.

Engine makers lay some stress upon having the indicator cards from the two cylinder ends look alike. Equality of cut-off has more influence on this than any other factor. For this reason and not because equality of cut-off is necessarily conducive

to smooth running, we will give a method* for finding the unequal laps and the angle between eccentric and crank when the cut-off is equalized for both strokes, say, made equal to 0.8.

In Fig. 64 take the distance OO' equal to the sum $e + e'$ of the laps at the two ends, and on stroke AB and $A'B'$ describe crank-pin circles. Strike off the equal and parallel arcs BK and $A'R$ with a radius equal to the connecting-rod, as in Fig. 63. Make $KL = RS = 0.8 \times \overline{AB}$. In the neighborhood of L draw a series of perpendiculars to OO' which cut the lower part of crank-pin circle BLA at points $u_1 \dots u_4$. Then with S as a center,

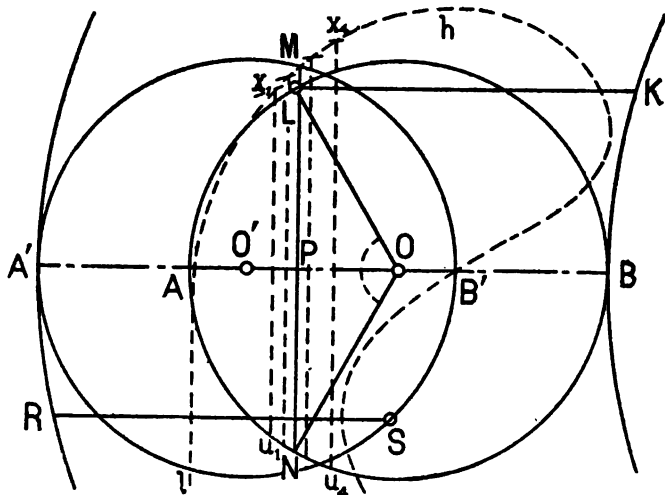


Fig.

and the distances, Lu_1, Lu_2, Lu_3, Lu_4 , etc., as radii, strike off the arcs that will cut the corresponding perpendiculars through u_1, u_2, u_3, u_4 , etc., at the points x_1, x_2, x_3, x'_4 . Join these intersections by a curve h . It will cut at M the other circle on $A'B'$.

From this point M drop another perpendicular MN to OO' cutting the latter in P and the lower part of circle ALB in N .

* See Burmester's "Lehrbuch der Kinematik," pp. 672-674, and Figs. 675 and 676.

Then will chord $LN = SM$ by construction and the angle $LON = \text{angle } SO'M$. Each of these angles is equal to the angle between crank and eccentric and hence to the angle between diameter of valve-circle and dead point line of eccentric (see p. 137). For this last angle $DOB = DOL + LOB$, Fig. 63, and the first part DOL has a cosine $= \text{lap} \div \text{eccentricity}$. The second part LOB is represented by LOP or $SO'P$ in Fig. 64 and the first part by PON and $FO'M$. Hence $PON + LOP = LON = PO'M + SO'P = SO'M = \text{angle between crank and eccentric}$. By giving the greater lap OP to the end of the valve farthest from shaft and lap $O'P$ to the other end of the valve the two cut-offs will each equal 0.8 and the eccentric setting $= LON$. The other intersections of auxiliary curve h with $A'MB'$ are of no use in this connection.

Dr. Burmester also solves another problem, namely, one in which, for a given setting of eccentric, the cut-off shall be equal. Construct circles on stroke AB and $A'B'$ with same center O and O' as before. Then take any point f in circle ALB and lay off angle fOe equal to given setting of eccentric. Through e draw ee' perpendicular to OO' , intersecting circle $A'MB'$ at e' . Lay

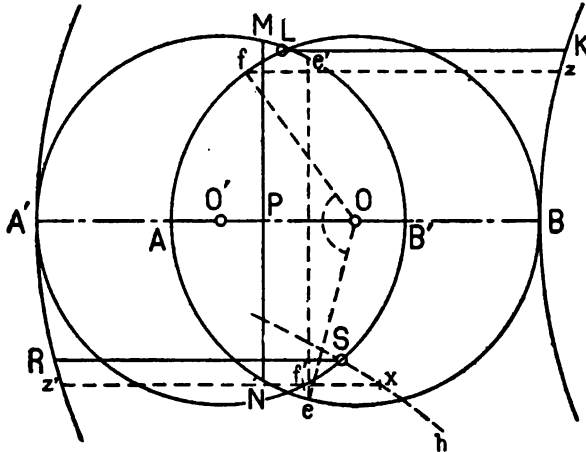


Fig. 65.

off $e'O'f'$ on this circle equal to the given setting and through f' and f draw parallels $z'f'$ and zf to OO' till they cut the arcs $A'R$ and BK . Then make $z'x = zf$; this will give one point x of an auxiliary curve h . By similar constructions other points of this curve h can be found. The auxiliary curve h cuts the circle $A'MB'$ in some point S . Finally make angle $SO'M = fOe$, drop the perpendicular MPN on to OO' and make angle $NOL = fOe$. Then will cut-off $\frac{RS}{A'B'} = \frac{KL}{AB}$, and OP will be the outside lap of the valve at the end farthest from the shaft and $O'P$ the lap for the other end of the valve.

In the mathematical discussions of valve-gears, it is found more convenient to use the angle of advance δ instead of the eccentric setting or angle between crank and eccentric. It may be expressed by a formula or it may be measured in any valve-gear as follows: Start with eccentric arm half-way between its own dead points and then move the crank to its (the crank's) nearest dead center. During this motion the angle passed through by either crank or eccentric will be the angle of advance and it will be positive if this motion or rotation is in the same direction as the engine rotation; otherwise it will be negative. The formula for this angle at any crank position is:

$$\delta = \lambda - \omega' - 90^\circ \quad (86)$$

Letting κ represent angle between crank and eccentric, we have

$$\kappa = 90^\circ + \delta \pm \chi = \lambda - \omega' + \chi. \quad (87)$$

where χ represents the angle between the two dead point lines and

* According to Rankine: "By angular advance is to be understood the angle at which the eccentric arm stands in advance of that position, which would bring the slide-valve to mid-stroke when the crank is at its dead points." λ represents the angle made by the eccentric with its own dead point, ω' the angle made by the crank with its dead point line; either of these angles may be greater than 180° .

$\left\{ \begin{array}{l} \text{upper} \\ \text{lower} \end{array} \right\}$ sign is used when dead point line of eccentric
 $\left\{ \begin{array}{l} \text{follows} \\ \text{precedes} \end{array} \right\}$ that of the crank in the direction of rotation.

When there is a reversing lever between eccentric and valve the formula for the angle between crank and eccentric then becomes

$$\kappa' = 180^\circ - \kappa = 90^\circ - \delta \mp \chi. \quad (88)$$

Sometimes a reducing lever is placed between the valve and eccentric for the purpose of reducing the size of the latter and thus diminishing its tendency to heat under high speed. The only modification which this makes in our diagram is to make the valve-circle diameter correspondingly larger. For we assume that the valve is driven directly by the eccentric. Sometimes, as in Corliss valve-gear, a reducing lever is used to give a sort of differential motion, and the valve direction and chord of lever-pin are made to differ purposely.*

The introduction of a *reversing* lever between eccentric and valve has the same effect on the valve motion, as if the eccentric were shifted to a diametrically opposite position and the valve then driven directly (*i. e.* without reversal) by the new eccentric. Hence here also, unless the contrary is specified, it is to be understood that diagram is drawn as if valve were driven directly by the eccentric.

This finishes our discussion of the simplest case of valve gearing. Under the head of single-valve gearing, with invariable steam distribution, there still remains a case which is mainly of interest because of its bearing on link motions, namely, the case in which the valve stroke does not pass through the center of the shaft, but at a certain distance c from the latter. In this mechanism the stroke of slide is more than twice the length of the eccentricity; moreover, the two dead points of the crank are not diametrically opposite as in the ordinary slider-crank.

* See also levers $O'RP$ and $O'RQ'$ Fig. 82, Porter-Allen Engine.

In Fig. 66 V is a valve, whose line of stroke VY passes center of shaft at distance $OY = c$. It is driven by an eccentric $OE = r$ and rod EV of length l . In valve-gears the point from which valve-travel is usually estimated is one near the middle of the valve stroke; the point of valve whose travel is measured may be any point rigidly attached to valve; in this figure the right hand end V of eccentric-rod is the one chosen. If we find the position of V corresponding to each of the crank's dead points and then bisect the distance between these two positions of V , the point of bisection will be the point from which valve-travel is measured.

Let M be this center of valve-travel and OE any position of the eccentric. MV will then be the corresponding valve-travel or distance from middle position. At the point E draw EC equal

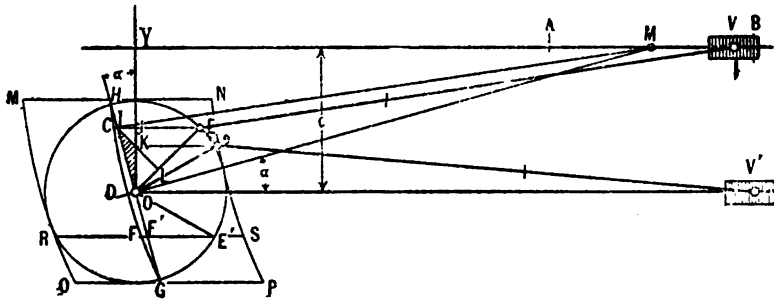


Fig. 66.

and parallel to MV , and join MC . The quadrilateral $CEVM$ is consequently a parallelogram and $CM = EV = l$. In other words, the locus of the point C is a circle described from M as a center, with radius l . The valve-travel for any other eccentric position OE' can therefore be found by drawing the parallel $E'F$ up to the arc GDH .* Draw the chord GOH of this arc per-

*The travel of the valve from the end of the stroke for any position OE' of eccentric is evidently either $E'S$ or $E'R$, according as one or the other end of the stroke is meant. The arcs MSO and NSP are struck, from the ends A and B of valve-stroke, with radius l of rod.

pendicular to line OM ; in this figure arc and chord meet at points G and H that seem to be on eccentric circle $EE'R$; but they do not necessarily lie on this circle.

Even in this sort of valve gears the rod l is long in comparison with the radius r , the rise OD of the arc is very small and the arc may then be replaced by the chord GH passing through center O . The distance EI , parallel to stroke, now represents the valve-travel, and for any other eccentric position OE' the distance from E' to the straight line GOH measures the travel of the valve. When eccentric center E and point H are on the same horizontal, the valve-travel EH will be the same whether measured up to chord GOH or arc GDH ; the same may be said when E and point G are on the same parallel to stroke. These two points of E , if constructed, would be found to be diametrically opposite and equidistant from arc or line of reference; if from each of these E 's as a center and with radius l we describe an arc, cutting stroke AB , and then bisect the distance between these arcs the point of bisection will be the center of reference M assumed above.

But the travel may be more easily found and this case reduced to that of the ordinary eccentric, by finding an eccentric Oe whose distance eK from the vertical OY is always equal to EI .* To find such an eccentric, we drop from point I a perpendicular IL upon the eccentric OE , and erect Ee perpendicular to OE . Now from the point K , where IL cuts the vertical Y , we draw Ke parallel to EI and join O and e . It is evident that $eEIK$ is a parallelogram and $eK = EI$. To be an equivalent eccentric Oe must not change its length or position relatively to OE , that is, $Ee = IK$ must be a constant for all positions of the original eccentric OE . The figure shows that triangle OKL is similar to

* Up to this point the demonstration is like that given by Prof. A. Fliegner in his work "Umsteuerungen der Locomotiven."

each of the triangles IJK and OJE . These triangles are therefore similar to each other, and we have

$$\frac{IK}{OE} = \frac{IJ}{OJ} = \text{tangent of angle } HOY = \frac{e}{l} \text{ nearly.} \quad (89)$$

But this angle is constant, being equal to the angle MOV , hence $IK = Ee$ is a constant, and Oe has all the properties of a virtual or equivalent eccentric. If we connect e by a rod l with a slide V' , whose stroke passes through O , the travel of V' from its middle position will be very nearly like that of the slide V from M . Evidently stroke of V' is greater than if driven by E .

The distance of E from chord GOH , or of e from vertical OY , does not exactly represent the travel of the valve, for it does not take account of the angularity of the eccentric rod (though it does consider the direction of line OM , and thus takes partial account of the length of this rod when distance e is given). The exact valve-travel is only given by the distance of eccentric-center E from arc GDH .

SINGLE-VALVE GEARS, STEAM DISTRIBUTION VARIABLE.

The characteristic feature of a single-valve gear is that its valve slides on a fixed, stationary seat (p. 134). In this sense an engine may contain one or more single-valve gears which divide among themselves the functions of the steam distribution.

The variable elements in these gears exist mainly to vary the expansion. They may roughly be arranged into two groups, those in which the driving eccentrics themselves can be varied, and those in which the eccentrics are themselves non-adjustable, the variation in the valve motion being effected by mechanism between the eccentrics and the valves. To the second group belong the link-motions, and these we will take up in the Appendix, because they are more complex and less extensively used in high-speed engine work than the first group.

In the first group the eccentric is varied by simultaneously changing both its throw and angles of advance.* The principal means by which this change is accomplished is the "swinging eccentric" device. In this the center of the eccentric is moved across the shaft in an arc **LL** having the point *P* as its center, as in Figs. 68 to 71 and Figs. 73 to 78. Ten of the "Single-Valve Automatics" represented use this device, the mechanisms for producing the swinging motion differing more or less; (see catalogues of engine builders). The object of this change is to alter the power of the engine by altering the amount of expansion, but other changes in the steam distribution also take place at the same time when there is but a single valve to effect them. For example, the compression begins earlier when the expansion begins earlier, great cushioning going hand in hand with great expansion. These two functions of the steam distribution have perhaps the most influence on the form of the indicator card and thus are the principal means of regulating the power per stroke and the uniformity of the driving force. These are important

* The only engine known to the writer that uses a single valve and varies the cut-off by varying only the angle of advance, is the oscillating engine built by the J. T. Case Co. Strictly speaking this engine has two valves, for the ports in the rocking cylinder control release, exhaust closure and admission, the cylinder itself performing the functions of the main valve.

On the other hand, the Buckeye engine has apparently a double-valve gear and yet in reality is made up of two single-valve gears the first being the one to which the main valve belongs and the second having the expansion valve. In this second gear, the expansion valve slides on a moving main valve, but the mechanism is such that the *relative* motion of expansion to main valve is scarcely affected by the *absolute* motion of main valve. The latter therefore acts as a stationary seat of the expansion valve which may be discussed and designed as if the ports it controlled were fixed and as if its rocker arm had a stationary pivot. We have however discussed the Buckeye's gear in Figs. 96-98, with the double-valve gears, which it so strongly resembles externally. When an engine has its distribution effected by several valves, each sliding on a fixed seat, the functions of one of them may well be varied by varying only the angle of advance of its eccentric or only the throw or lap.

factors in securing *steady* running of the engine. To effect *smooth* running there must be not only absence of great fluctuations of power per revolution, but also absence of vibrations and shocks. Shocks occur when there is "play" (or clearance) at a bearing accompanied by such sudden and complete changes of direction in the force acting on the bearing that continuous contact between the pin (or slide) and its bearing ceases while the space (or "play") between the two is traversed by one or the other of the two pieces. Tremors and vibrations of course accompany shocks, but they may also arise independently, from sudden changes in the intensity of the pressure between two pieces. Thus at the end of a piston stroke the final cushion-pressure may be considerably less than the initial pressure on piston at beginning of next stroke, the sudden, additional, load on piston sending a tremor through all the connections. Sudden reversals of pressure should therefore be avoided, particularly at the dead point where they are most dangerous, (see Appendix). As the extent to which cushioning is carried, the character of the lead (positive or negative) and its extent, strongly influence these dead-point-pressures, the variations effected by the valve gear in these quantities per revolution and at the different grades of expansion becomes a matter of consequence. In times past engineers laid great stress on having the steam lead equal at the two cylinder ends whatever the grade of expansion, particular care being taken to "set" the valve with equal lead. This desire for *equalization* at the two cylinder ends extended itself to the cut-off, the release and the cushioning as well as to the lead. But it was recognized, particularly in locomotive practice, that it was impossible with existing valve gear to *equalise* all the functions at the same time, equalization of any one function was at the expense of the others. Generally it was the lead (a positive one) that was thus favored. At the same time most engineers also held that a lead that was constant at all grades of expansion was a desideratum. It was regarded as the special merit of the Gooch link-motion that it

possessed this virtue. But since the advent of the high speed steam engine a difference of opinion on this matter has developed itself among engineers.

In the pioneer engine of this type, the Porter-Allen, the leads were deliberately made unequal at the two ends (Figs. 86, 87 and Figs. 89, 90 of Porter-Allen valve diagrams) and the lead also varied from mid-gear to full gear, as in other link-motions. The Straight Line engine in its earliest forms had a lead that was equalized at the ends and was nearly constant at all grades of expansion; in the later forms the lead is now variable for different cut-offs; it is sometimes negative and there is some inequality at the ends. In other and excellent engines the old views on this subject are still carried out so that the matter cannot be regarded as finally settled either way, Prof. Sweet, the designer of the "Straight Line" engine, going so far as to determine experimentally, with the indicator, the conditions of smooth running for each engine. The collection of examples given shows the variation of American practice in this respect. We shall not attempt any comparisons of these engines, for the exact effect of the steam distribution on the steadiness and smoothness of running can only be determined by a knowledge of the clearances, the pressures, the weights of the reciprocating parts and the regulating capacity of the governor under variable loads.

We have thus far said nothing concerning the release, because it has usually less influence on smoothness of running than the other factors, though it too effects somewhat the pressure on piston at dead point. The release is of consequence to economical running because of its influence on the amount of back pressure, an early release and widely opened exhaust ports tending to keep this pressure small. On the other hand release begins too early when it curtails the period of expansion to any notable degree. We have given the beginning of the release for only a few cases; it may be easily found from the exact diagrams, by following the directions given with Fig. 67.

Returning now to the influence that the shape and location of the locus LL of the eccentric centers E has on the steam distribution, we note first, that as the chord of the utilized portion of the arc LL is more nearly perpendicular to the dead point line of the eccentric, the lead is more nearly constant at the different grades of expansion, provided the curvature is the same, and secondly, that with this perpendicular location an arc LL of great radius (*i. e.*, of slight curvature) causes smaller variations in the lead than an arc described with a small radius.

Radii drawn from the center O to the locus LL (or $E_oE_1E_2$) of centers, give the length and position of the eccentrics corresponding to the different grades of expansion, the locus moving with the crank to which it is rigidly attached. The broken line $D_oD_1D_2$ is the locus of the vertices of the diameters of the valve circles corresponding to the eccentrics E_o, E_1, E_2 . The position of these diameters were found from their eccentric by laying off the angle σ as in Fig. 60. In Fig. 67 the locus $L'L'$ or $E_{v_0}E_{v_1}E_{v_2}$ was derived from LL , the locus actually described by the swinging eccentric, by constructing a series of offsets each equal to $\frac{1.5}{34} r_n$, according to the method given in connection with Fig. 66.

The new eccentrics thus obtained are regarded as virtual eccentrics and their centers are indicated by $E_{v_0}, E_{v_1}, E_{v_2}$. The valve circles, were derived from the virtual eccentrics, $OE_{v_0}, OE_{v_1}, OE_{v_2}$.

The particular eccentrics chosen for representation are those corresponding to minimum, quarter and maximum cut-off, the first and last of these being taken from data furnished, in most cases, by the manufacturers themselves. The quarter cut-off is generally an average value of the cut-offs in the two cylinder ends when angularity of connecting rod is taken into account. The intermediate cut-offs in the case of the Westinghouse Engines are somewhat larger than 0.25 or about 0.30. Each of these valve circles will give the steam distribution shown in Fig. 63, when the lap and port-width circles are drawn. The crank

positions and piston travel at cut-off, preadmission and beginning of compression are only drawn for the eccentric and valve circle corresponding to $\frac{1}{4}$ cut-off; the shaded area shown in the diagram cuts, from the crank, intercepts that represent the port-openings at quarter cut off. If there is more than one port each intercept must be multiplied by the number of ports to get the total opening. It must not be forgotten that each of these valve-circles cuts from the crank or its prolongation a chord that represents the distance of the eccentric center E from the perpendicular YY , through O , to the dead center line of the eccentric. With the exception of Figs. 74, 77, this perpendicular is the vertical through O . The valve circles entirely neglect the angularity of the eccentric rod. Whenever the length of the latter was known, a dotted, central, arc of reference was drawn from which the valve travel from the center of this arc can be found exactly. This arc center is obtained as follows: the crank is placed at one of the dead points and the center E of the eccentric in question (usually the one for maximum cut-off) is placed in its corresponding position, then with E as a center and length of eccentric rod as a radius strike off an arc cutting the valve stroke; repeat the operation for the other dead center of crank, the point bisecting the distance between these arcs will be the center of the arc of reference required. In the Rice, Ball, Southwark, Straight Line and Westinghouse Compound engines, the central arc of reference differs so little from the vertical through O that no effort was made to draw the polar curves representing exactly the valve travel, the valve-circle being sufficiently accurate for all practical purposes. In the Westinghouse Standard there is also only a slight difference between the perpendicular $Y'Y'$ and the arc $Y'Y'$; still for the case of maximum eccentricity the exact polar curves were plotted on the diagram in dotted lines, the one outside the valve-circle corresponding to the positions of the valve to the left of the center of the reference arc and the one inside to the positions of the valve to the right of this center.

Comparison with the intermediate valve-circle shows that the steam distribution is but slightly changed by substituting the exact polar curves for the less exact circle.

The cut-offs are apparent ones and are estimated from *beginning* of stroke in the same manner as in former sections. Pre-admission, beginning of compression and release are however measured from the *end* of the stroke and each is then divided by the stroke itself, so that the results are always expressed as fractions of the stroke. The release is not tabulated on the diagrams. The symbols used are

a = width of port.

l = length of port in cylinder.

d = diagram of piston valve.

e = steam lap.

i = exhaust lap.

ε = apparent cut-off.

v = steam lead.

r = eccentricity.

R = radius of crank.

L = length of connecting-rod.

ε' = beginning of release, measured from end of stroke and divided by stroke.

η = beginning of compression, measured from end of stroke and divided by stroke.

η' = beginning of admission, measured from end of stroke and divided by stroke.

The period of compression would properly be represented by the difference $\eta - \eta'$.

The subscripts ₀, ₁ and ₂ affixed to these symbols signify that they belong to minimum, quarter and maximum cut-offs respectively, as explained above.

All the engines represented "run over" (see p. 118), *i.e.*, in these diagrams, the rotation is right-handed. Then in Fig. 67, for the distribution corresponding to quarter cut-off, KT repre-

sents admission of steam to head end during forward stroke and SR to crank end for return stroke. FH represents compression in crank end during forward stroke and GU compression in head end during return stroke because exhaust lap is negative. $p_1 q_1$ represents the preadmission to head end before beginning of forward stroke and $p q$ the preadmission to crank end of cylinder for return stroke. As the inside lap at crank end is different from that at head end we must carefully distinguish between the two. Then $m_1 n_1$ is that last part of the forward stroke during which release takes place in the head end and mn that part of the return stroke during which release takes place in the crank end of the cylinder. By dividing these quantities by $2 \times \overline{CO}$, we get the values tabulated in the diagrams. We have omitted the release from these tabulations but it can be easily found from these exact diagrams if desired. In following the steam distribution by means of the diagram, it is well to bear in mind that the valve is to the right of its middle position whenever the valve-circle cuts the crank radius.

The same letters are used in the other diagrams to represent the same parts of the distribution; the letter C always represents the crank position, E_* eccentric position corresponding to C , and D_* the diameter of the valve-circle corresponding to the eccentric-setting COE_* .

The valve diagrams, for the twelve Single-Valve Automatics represented, are here divided into two groups according to the location of the pivot P (or center of curvature) relatively to the crank radius. In the first group, Fig. 67-72, the center of curvature of that element of the locus LL which crosses the crank lies on the crank's own radius, while in the second group, Figs. 73-78, the pivot P lies outside of the crank's radius or its prolongation. As the pivots P are often at considerable distance from their loci LL , economy of space required that the points be detached from their proper position relatively to their loci and be placed nearer the latter; this has been done in every case and their co-ordinates given so that they can be laid out if desired.

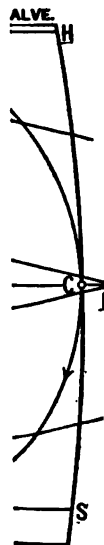


AD END

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AD END

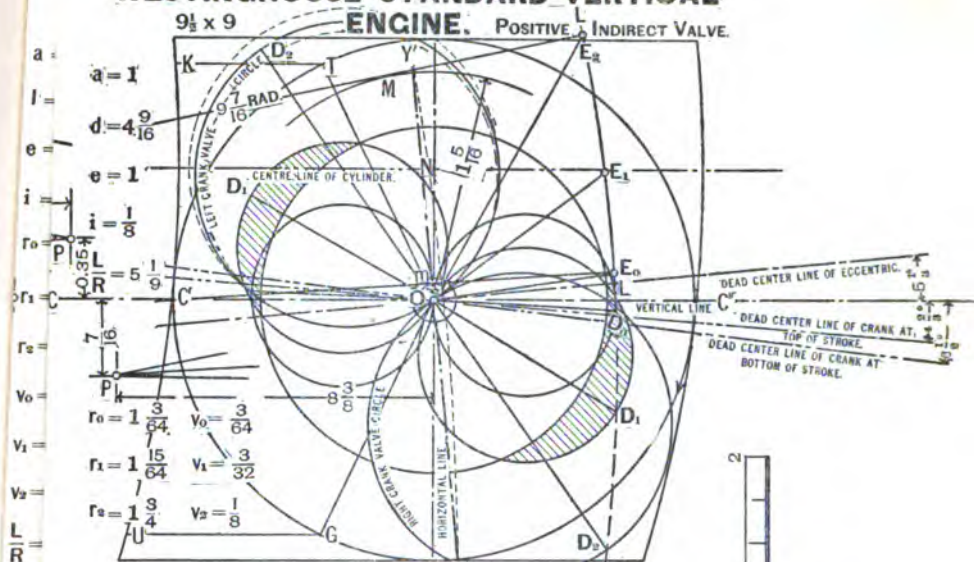
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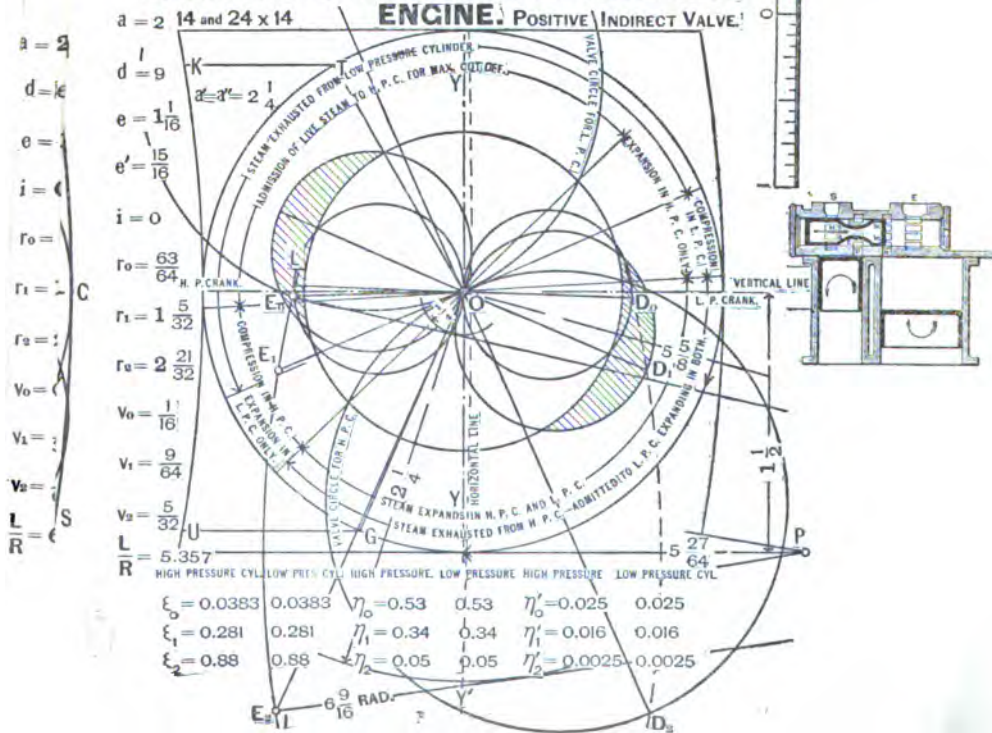
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Figs. 77 and 78.

WESTINGHOUSE STANDARD VERTICAL ENGINE. POSITIVE INDIRECT VALVE.



WESTINGHOUSE COMPOUND VERTICAL ENGINE. POSITIVE INDIRECT VALVE.



RUSSELL & CO. ENGINE, 14X20.

Fig. 67. In this valve gear the center of the eccentric is moved in a perfectly straight line LL across the shaft, at right angles to the crank. The mechanism for accomplishing this (it is not the "swinging eccentric" device) is not given here but can be found in catalogue of the builder. Inspection of the valve-circles shows at once that the steam lead v is constant at all grades of expansion. The lead on the exhaust side is however unequal at the two cylinder ends, for the inside lap $i = 0$ and $-\frac{1}{8}$ at the crank and head end respectively. But this is not a matter of any consequence. The inequality in the inside lap equalizes the beginning of release and of compression for the two cylinder ends as the following table shows:

	Crank End.	Head End.		Crank End.	Head End.
ϵ_0'	0.54	0.54	η_0	0.458	0.450
ϵ_1'	0.36	0.37	η_1	0.292	0.284
ϵ_2'	0.075	0.075	η_2	0.065	0.050

The value of η includes the preadmission of steam and so does not exactly represent compression proper. But it may be taken as representing the *cushioning*, which is practically equalized for the two ends by this arrangement. The table connected with the diagram gives the variation of cut-off and preadmission for minimum, average and maximum throw; the range of cut-off varies from $\frac{1}{11}$ to $\frac{8}{10}$. This engine uses the Giddings' Balanced Valve, which is divided into two parts, rigidly connected, one part for each end of the cylinder. Each part admits steam through two passages that run through the valve and somewhat resemble the supplementary passage of the well-known Allen valve. Like the latter this valve belongs to the positive direct type and increases the clearance volume during the compression period by the volume of the supplementary passage. The clearance volume is smaller at the beginning than at the end of the

compression. As there are two ports or passages for each end of the cylinder, the admission and cut-off are prompt, and the port opening given by the shaded area of the diagram must be multiplied by 2. The valve is of the flat and balanced variety. The speed of this size of engine is 210 revolutions per minute.

N. Y. S. S. POWER CO., ENGINE, 12x12.

Fig. 68. The locus **LL** of the center of eccentric is concave to the shaft *O*. The center of this arc is on crank radius, at pivot *P*, and 14" from center *O* of shaft. The lead diminishes as the cut-off increases. The cut-off ranges from $\frac{1}{11}$ to $\frac{4}{8}$. The beginning of release is given by this table,

	Crank End.	Head End.
ϵ_0'	0.54	0.458
ϵ_1'	0.35	0.276
ϵ_2'	0.065	0.046

PAYNE ENGINE, 13x12.

Fig. 69. The location of the pivot of the pendulum arm of the swinging eccentric in the smallest engines is different from that here given. In a 5x7 for instance, the pivot *P* and crank-pin *C* are on opposite sides of center *O*; in this case pivot *P* and pin *C* are on the same side of *O*. The pivot is $11\frac{1}{2}$ " from shaft center *O* and lies on the same radial line as crank. The radius of arc **LL** is $10\frac{5}{8}$ "; this makes the arc convex to the shaft and the lead increases with the cut-off, the latter ranging from 0 to $\frac{3}{4}$. The fraction of piston stroke occupied by preadmission is very small. The valve takes steam on the inside and is of the flat, balanced, variety. This size of engine makes about 250 revolutions per minute.

RICE ENGINE, 7x10.

Fig. 70. This engine presents a unique appearance in that no governor is visible. The regulator is within the crank disc and

turns an arbor that passes through the center of the crank pin C . This arbor is the pivot P of the pendulum arm that swings the eccentric-pin across the shaft when the expansion is to be varied. The eccentric-pin and the pendulum arm constitute a sort of return crank. In the figure, both C and P represent the center of pivot and of crank-pin. The locus LL is again convex to the shaft and has a nearly constant lead; its radius is $4\frac{1}{2}$. The fraction of the stroke occupied by the preadmission is larger in this engine than in any other of the dozen represented. The cut-off ranges from $\frac{1}{4}$ to $\frac{7}{10}$. The length of the eccentric rod is only $15''$, but the influence of its angularity on the valve travel is but slight, as the arc $Y'Y'$ shows. This arc was struck from the center of motion corresponding to maximum throw and with a radius of $15''$. (The center of motion of valve is half way between the two valve positions corresponding to the crank's two dead points). The vectors of the valve circle represent the distance of point E_* from vertical YY and are very nearly equal to the valve travel from the center of motion. The exact valve travel at any instant is given by the distance of point E_* from the arc $Y'Y'$. These exact travels could be laid off on the crank positions or their prolongations and thus two exact, polar, diagrams be found (for each eccentric throw) which would cut from the crank the exact distance of the valve from the middle position. But in this case they would differ so little from the intermediate valve circle that this nicety of construction has been omitted. The valve is of the flat, balanced, variety.

BALL ENGINE—OLDER TYPE, 9X12.

Fig. 71. The diagram of this engine is given mainly for comparison with the diagram of a more recent style of valve-gear for the same engine whose diagram is given in Fig. 76. In this earlier form the point P of the swinging eccentric was $13\frac{1}{2}''$ distant from the center of the shaft O and was on the same radial line as the crank-pin C . The locus LL of the center of the eccentric was convex to shaft O and gave a lead that increased

as the cut-off increased. The range of cut-off was from 0 to $\frac{5}{8}$. The valve was flat and balanced. The length of the eccentric rod was about $41\frac{1}{2}$ inches and with this as a radius an arc $Y'Y'$ is struck off from the center of motion, in the manner already explained when discussing the Rice Engine; the distance of the center E_e of eccentric from dotted arc $Y'Y'$ gives the exact travel of the valve, while the valve circle gives the distance of E_e from vertical YOY ; here again the difference is so slight that we may rest well content with the results furnished by the valve circles.

ARMINGTON AND SIMS ENGINE, $14 \times 15\frac{1}{2}$.

Fig. 72. In this engine a peculiar linkage is employed to vary the position of the center E of the eccentric; for the details of this linkage we must refer the reader to the catalogue of the engine builder. The locus LL of this diagram is not a circular arc though it closely resembles some of the loci that have already been given. It was plotted from an accurately measured linkage belonging to a $14 \times 15\frac{1}{2}$ engine. Here also, the lead increases as the cut-off increases, the table on the diagram showing the extent to which this takes place. The range of cut-off is from 0 to $\frac{3}{8}$. As in all the preceding cases the compression or cushioning period increases with the period of expansion. The eccentric rod which connects the driving eccentric with the $10\frac{1}{4}$ " rocker arm is itself 44" long. We may regard the rocker arc as a straight line and repeat the construction of the arc $Y'Y'$ already given with the Rice & Ball engines. Here also the difference between vertical reference line YY and the reference arc $Y'Y'$ is so slight that it may be neglected. The valve is of the hollow piston variety, is $6\frac{1}{4}$ " in diameter, is of the positive indirect type and takes steam simultaneously at two places. One of its steam passages leads through the center of the valve and in principle is like the supplementary passage of the Allen valve that is still used on some locomotives. This size of engine is run at about 280 revolutions per minute.

SOUTHWARK ENGINE, 9X10.

Fig. 73. It is the first of the second group of Single-Valve Automatics. In this group the locus LL of the driving pin or eccentric E of the valve has its center P outside of the crank radius OC . In most of these cases the pivot P and crank-pin C are still both on same side of shaft center O . In the present case the co-ordinates of P are $5\frac{1}{4}''$ and $-\frac{5}{8}''$ which brings it opposite the middle of the utilized portion of the arc LL . The arc is concave to O and its chord is perpendicular to crank radius so that it gives a nearly constant lead, for all grades of expansion. The cut-off ranges from 0 to $\frac{2}{3}$ and the cushioning from $\frac{2}{18}$ to $\frac{1}{7}$. This valve-gear and its governor were designed by Prof. C. B. Richards. The substitution of a pin E in place of the eccentric sheave is an excellent feature as it removes the danger of overheating to which the eccentrics of high-speed engines are so liable.

The rod coupling the driving pin E with the $3\frac{1}{4}''$ rocker is $34''$ long; treating the rocker arc as approximately a straight line, we can repeat the constructions of the arc $Y'Y'$ already given with the three preceding engines. Here too the deviation of the more exact reference arc $Y'Y'$ from the vertical reference line YY is so slight that there is no need of substituting exact polar curves for the valve-circles. The valve used in this engine is flat, balanced, and of the positive direct type. The speed of this 9x10 engine is 300 revolutions per minute.

THE STRAIGHT LINE ENGINE, 11X14.

Fig. 74. The locus LL is concave to O and has the point P outside of the crank radius OC , but P and C are both on the same side of O , the abscissa of P being $12.25''$ and its ordinate $-2\frac{1}{2}''$. The stroke of the valve does not pass through the center O of the shaft, but passes it at a distance of $1\frac{1}{2}$ inches. The dead point line of eccentric will therefore be different for the two ends of

the valve strokes and for the different throws of the valve. An average position has been drawn on the diagram. It is therefore like the case represented in Fig. 66, p. 151. The offset Ee is there shown to be nearly equal to $\frac{c}{l}r$; here $c = 1.5''$, $l = 34''$ and $r =$ eccentricity corresponding to grade of expansion considered. If $r = 2.5''$, then $\frac{1.5}{34} \times 2.5 = 0.11$ is the offset, which laid off from E_2 at right angles to OE_2 gives E_{v_2} which is the center of the virtual eccentric OE_{v_2} . In like manner other offsets may be found for other throws of the valve and thus a curve $L'L'$, passing through the ends of these offsets, can be found which will be the locus of the centers E_v of the virtual eccentrics OE_v . The valve-circle belonging to any eccentric can be found as in Fig. 60 and in either of two ways: it can be obtained from the actual eccentric (drawn from O to locus LL) by laying off the angle σ from the dead point line of eccentric, or it can be obtained from the virtual eccentric OE_v by laying off an angle σ from the dead point line of crank. (See p. 137, Fig. 60 and Fig. 66, p. 151). The chord cut from the crank by the valve circle thus determined equally represents the horizontal distance of E_v from the YY line through O or the horizontal distance of E_{v_n} from the vertical line $Y'Y'$. (The chords of the valve circle do not represent either of these distances exactly on account of an approximation made in the construction, Fig. 66.) In neither of these determinations is the angularity of the eccentric-rod fully taken into account. To do this we proceed as in preceding cases. Thus from E_2 as a center, with length of eccentric-rod ($= 34''$) as a radius we strike an arc that cuts the central line or stroke of the valve; with same radius and E_2' as a center a second arc is struck off, again cutting the line of stroke of the valve; the point on the stroke bisecting the distance between these arcs will be the center of motion, from which as a center and $34''$ as a radius the arc of reference $Y''Y''$ can be described. The distance of E_v from this arc gives with great exactness the actual travel of the valve

from the center of motion for the assumed eccentric OE_* . In this case, the arc of reference $Y''Y''$ does not hug the line YY so closely that it is evident that the vectors of the valve-circles represent with sufficient accuracy the travel of the valve. We therefore give in the following table the *exact* steam distribution for $r_2 = 2.5$ and $r_1 = 1.48$, the latter value differing a little from the value $r_1 = 1.41$ assumed in diagram. The following values were found by constructing the exact polar curves.

Distri- bution.	$r_1 = 1.48.$		Distri- bution.	$r_2 = 2.50.$	
	Crank End.	Head End.		Crank End.	Head End.
ε_1	0.31	0.29	ε_2	0.71	0.77
ε'_1	0.10	0.12	ε'_2	0.04	0.04
η_1	0.34	0.34	η_2	0.11	0.13
η'_1	+ 0.00	- 0.005	η'_2	+ 0.00	+ 0.00

This is almost perfect equalization of the distribution for the two ends. A careful comparison of these results with those given in the diagram show that, with perhaps the exception of cut-offs near $\frac{1}{4}$, the distribution is given with sufficient accuracy by the valve circles even in this extreme case.

The numerical values inscribed on the diagram show that cut-off ranges from about 0 to $\frac{3}{4}$, that the cushioning is nearly equal for the two ends of the cylinder and that the lead varies from a negative value of 0.11 at minimum throw to positive value of 0.11 at maximum throw. The lead is known to be about equal at the two ends for this throw. But this preadmission is attained in very small fractions of the stroke as the tabulated values η' show.

The minimum throw chosen was that one in which the port is opened only for an instant. The almost perfect equalization of the functions of the exhaust illustrates the value for this purpose of making the laps unequal. There seems to be more clearance at head end than at crank end in this engine.

Steam is admitted at two places, the port openings given by shaded area of the diagram must therefore be doubled in order to get the true area available for the passage of the steam. The speed of this size of engine is about 275 revolutions per minute.

The Straight Line Engine was designed by Prof. John E. Sweet and constitutes a most instructive example in steam distribution. I am indebted to Prof. Sweet for the following outline of his practice in securing smooth running engines.

Originally the valve motion was designed to produce a practically constant lead at each end of the cylinder. Later experience convinced Prof. Sweet that a constant lead was exactly what he did not want and he therefore located the eccentric plate pivot *P* so that it would give a variable lead equally at the two ends when a correcting, rocker, arm device was used. Still later he found that the variable lead enabled him to dispense with the correcting arrangement, the lead at the two ends being then somewhat unlike. Prof. Sweet's practice on the testing floor is as follows: the first valve is made too long and with too much inside lap, then with varying loads and the indicator and the exercise of judgment as to smooth running, the valve is altered and tried until the best average is obtained. So much discrepancy has been found between different size engines of the same style and sometimes different engines of the same size that no attention whatever is paid to drawing room figures on the valve or its setting. One reason for this is the possible, disturbing, influence of the governor. Prof. Sweet prefers an increase of compression to an increase of lead at the head end of the cylinder; the compression will show more on the cards, but this is held to be of no practical account as the object is to obtain a smooth running engine rather than good looking cards.

In the smooth running engine for which the diagram was drawn, the distribution was practically equalized for the two ends of the cylinder. So far as this one example goes it would therefore seem that uniformity of distribution is favorable to smoothness of running.

STURTEVANT ENGINE, 8x12.

Fig. 75. The point P of the pendulum arm lies outside of the crank radius OC , but P and C are both on same side of shaft center O . The locus LL , struck from P as a center, is convex to center O of its shaft, but it is so located that the lead increases with the throw, and the preadmission takes but a small part of the stroke. The cut-off varies from $\frac{1}{4}$ to $\frac{3}{4}$. The piston valve has a diameter of 3" and is of the positive direct-type. The steam passages in the cylinder have a cross-section $\frac{7}{8}" \times 6"$. The speed of this engine is 250 revolutions per minute.

BALL ENGINE.—RECENT TYPE, 12x12.

Fig. 76. This is the diagram of a more recent type of Ball engine than that discussed in Fig. 71. Here the point P is outside of crank direction OC and P and C are on *opposite* sides of shaft center O . The locus LL is concave to O , but so placed as to permit but little change in the lead as the expansion varies; the preadmission also occupies but little of the piston stroke. The cut-off ranges from 0 to $\frac{3}{4}$. In this engine, as in the Southwark, a pin takes the place of the eccentric sheave, thus affecting a reduction of the work of friction and removing the danger of heating. The valve is of the flat and balanced variety; it exhausts at the outer edges and takes steam at the inner ones.

WESTINGHOUSE STANDARD VERTICAL ENGINE, 9 $\frac{1}{2}$ x9.

Fig. 77. In this diagram the left-cylinder-crank C' is at the left and the right one C'' at the right, both being placed on the vertical line of the engine (not of the diagram). The relative positions of crank and eccentric are as they would appear if viewed from the right-crank end of shaft. The terms right and left cylinder as used here and as used by the manufacturers supposes the engine viewed from the throttle side; then it is the $\left\{ \begin{array}{l} \text{upper} \\ \text{lower} \end{array} \right\}$ port that admits steam to the $\left\{ \begin{array}{l} \text{right} \\ \text{left} \end{array} \right\}$ cylinder. The

center line of each cylinder passes the center O of shaft at a distance equal to half the length of crank OC'' . In the diagram this distance and crank are drawn to a reduced scale.

The pivot P of swinging eccentric is $\frac{7}{8}''$ from the radial line of the left crank and has a negative abscissa of $8\frac{3}{8}''$. The radius of locus LL is $PE_2 = 9\frac{7}{8}''$. At maximum cut-off the eccentricity is $1\frac{3}{4}''$ and the radius PME_2 is tangent to the arc M described with $OM = 1\frac{5}{8}''$ as a radius; at minimum cut-off the eccentricity is $1\frac{3}{8}'' = OE_0$ and the radial line PmE_0 is tangent to the arc m , which is struck off with radius $Om = \frac{3}{8}''$. The valve circles are found as in Fig. 60, p. 137. The lead v , for all grades of expansion, was measured on that dead center line of crank which corresponds to uppermost position of piston and increases with the cut-off. The engine is composed of two single-acting, vertical, cylinders which receive the live steam in the top end of the cylinder. As each cylinder has its own crank and connecting-rod this arrangement will cause the angularity of the rod to have precisely the same influence on the steam distribution of each cylinder. This is shown by the values tabulated on the diagram. The cut-offs evidently range from 0 to $\frac{3}{4}$.

The table inscribed on the diagram and valve-circles (in full lines) do not take into account the influence of the length of the eccentric rod. The vectors of the valve-circles drawn from O give the hor. distance of the eccentric center E_2 from the line $Y'OY'$, which is perpendicular to the dead center line of eccentric. To take account of this angularity of the eccentric rod, we make the following construction. Place the crank upon dead point for top of stroke and the eccentric in a corresponding position. The center E_2 , will then be $4\frac{3}{8}''$ in advance of position shown in the diagram. Taking this new position of E_2 as a center and the $20''$ length of the eccentric as a radius, strike an arc that will cut the dead point line of eccentric; then revolve this eccentric through 180° and use the new position of E_2 as a center to repeat the former operation. Now bisect the distance between

these two arcs and the point of bisection will be the center of motion to which the valve travel is referred and from which as a center the arc of reference $Y'Y'$ can be described with the 20'' radius. The hor. distance of the eccentric center E_* from this arc of reference will be the exact travel of valve. If this exact travel is laid off on the corresponding crank positions, laying off valve travel to $\left\{ \begin{smallmatrix} \text{right} \\ \text{left} \end{smallmatrix} \right\}$ of center of motion on $\left\{ \begin{smallmatrix} \text{actual} \\ \text{prolonged} \end{smallmatrix} \right\}$ crank, repeating this process for second crank also, we will get two pairs of polar curves instead of the two valve-circles, and the intersection of these curves with the proper valve-circles will give the exact steam distribution. One pair of these polar curves, the one for left cylinder crank has been roughly drawn on the diagram for maximum throw. They lie on opposite sides of valve-circle OD_* which may be regarded as their average valve. The $\left\{ \begin{smallmatrix} \text{outer} \\ \text{inner} \end{smallmatrix} \right\}$ one of the polar curves represents travel to the $\left\{ \begin{smallmatrix} \text{left} \\ \text{right} \end{smallmatrix} \right\}$ of the center of motion. The exact steam distribution given by carefully drawn polar curves is contained in the following tabulation:

Half Throw.	Right Cylinder.				
$r_1 = 1\frac{15}{16}$	$\varepsilon_1 = 0.34$	$\varepsilon_1' = 0.21$	$\eta_1 = 0.38$	$\eta_1' = 0.01$	
$r_2 = 1\frac{3}{4}$	$\varepsilon_2 = 0.68$	$\varepsilon_2' = 0.08$	$\eta_2 = 0.16$	$\eta_2' = 0.00$	
Half Throw.	Left Cylinder.				
$r_1 = 1\frac{15}{16}$	$\varepsilon_1 = 0.28$	$\varepsilon_1' = 0.21$	$\eta_1 = 0.35$	$\eta_1' = 0.00$	
$r_2 = 1\frac{3}{4}$	$\varepsilon_2 = 0.67$	$\varepsilon_2' = 0.06$	$\eta_2 = 0.17$	$\eta_2' = 0.00$	

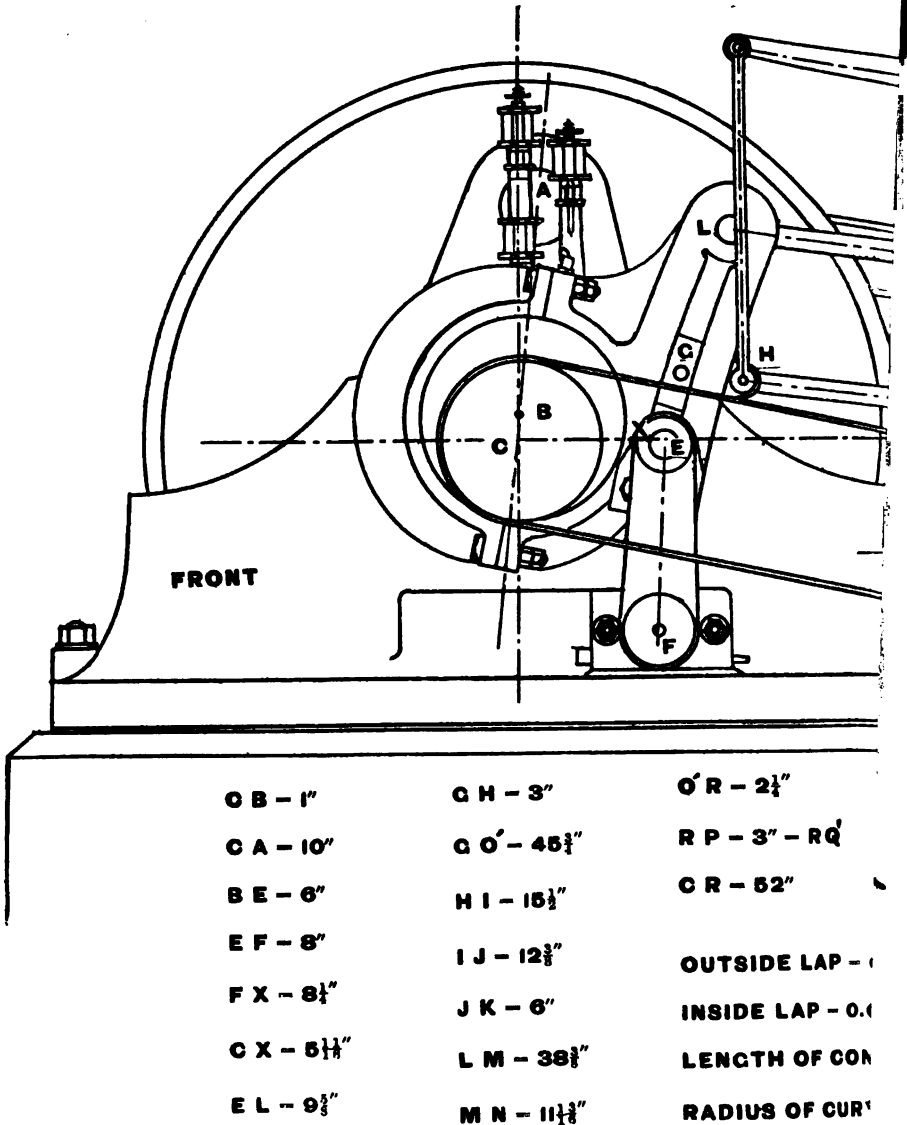
The slight inequalities that exist for the two cylinders is wholly due to the influence of the length of the eccentric-rod. The diameter of the piston valve is $4\frac{9}{16}$ '' and the speed is 300 revolutions per minute.

WESTINGHOUSE COMPOUND VERTICAL ENGINE, 14 AND 24X14.

Fig. 78. In this case the stroke of the pistons passes directly through the center of shaft. The single, piston, valve controls two ports M and N (see little figure to right of diagram 78.) The passage P leading to high pressure cylinder is always in communication with the hollow of the valve. The port M therefore admits live steam from pipe S to the high-pressure cylinder, and port N , at one edge receives steam from high pressure cylinder and admits it to the low pressure cylinder, and at the other edge it delivers steam from low pressure cylinder to the exhaust pipe E . The steam laps e at the inner edges are alike for both ports M and N . The exhaust lap is however equal to 0. The value of e' given in table was obtained by multiplying actual lap e by $\frac{7}{8}$ ($= \frac{2}{1.04}$, the ratio of the arms of the bent lever driving the valve) to reduce it and the stroke of the valve to the end of the eccentric rod. This is a departure from our usual practice of assuming that the actual valve is driven directly by the eccentric, but in this case it was simpler to change only one dimension than to change several.

The pivot P does not lie on either of the two cranks, but it is near to the low-pressure crank and on the same side of center O . The abscissa and ordinate of P are 5.42'' and 1.5'' respectively, its total, radial, distance from O being $5\frac{5}{8}$ '. The locus LL struck from this center has a radius of $6\frac{2}{16}$ ' and is concave to O . The table and data inscribed on diagram state distribution. In this Compound, as in the Standard, the steam distribution is the same for both single acting cylinders when the eccentric rod is taken as infinitely long. In this case the rod is finite and 49'' long and its influence on the distribution can be ascertained in the now well-known manner by drawing the arc of reference $Y'Y'$. It is evident from the figure and from what has been said about other diagrams, that the valve-circles give the valve-travel with sufficient accuracy.

Fig. 79-



LINK-MOTIONS.

In the second group of single-valve gears (see p. 153), the eccentrics themselves are invariable; the variation in the valve motion being affected by the mechanism between the eccentrics and the valves. All the common link motions belong to this group; we will here describe and examine but one of them, the Porter-Allen, (or 'Fink'), the only one which in this country has been extensively used on stationary, high-speed, engines.*

The Porter-Allen is that special case of the Gooch link motion which is obtained by supposing the centers of the two eccentrics to coincide, thus forming only one eccentric, the two eccentric rods and link then constituting one rigid piece. The best proportions for this linkage are shown in Figs. 79-82, and were taken from blue prints furnished by the Southwark Foundry & Machine Co. of Philadelphia.

PORTER-ALLEN VALVE-GEAR, 11½x20.

Figs. 79, 80 and 81 gives a side view of this engine, a section through its rocker arm and a skeleton showing the relative position of the three valves when crank is on its forward center. This view and the skeleton of the mechanism given in Fig. 82 are sufficiently full and complete to render any detailed description unnecessary. We will only ask the reader to note that eccentric strap and link constitute one rigid piece, that for a particular grade of expansion we assume that the governor occupies an invariable position vertically, thus making point of suspension /

* "Prof. Zeuner's views of the unsuitableness of the 'Fink' link as a *reversing-gear* are correct, and for the reasons he gives. But the use of the half-link in the Porter-Allen engines (it is in connection with these engines that the Fink motion was first brought into extensive use) shows that when compression is judiciously applied, then the cut-off points may be made practically 'symmetrical' up to the half-stroke *at the expense of a variable lead*, the variation of which is in opposite directions for the two strokes. Before cushioning was used in these engines the variability of the lead made it almost impossible to obtain quiet running under varying conditions."

of the hanger IH a fixed stationary point, and finally that even then the center G of block does not maintain an invariable distance from the trunnion E , but slips a little in its slot.

The view of the relative position of the valves is only a skeleton and does not represent the multiple port-openings; each of the three valves admits steam at four different places thus offering a generous passage way for both the live and exhaust steam. Before taking leave of these general views we desire to call special attention to the rocker arms $O'RPQ'$ which communicate motion to the rods and stems of the two steam valves. From the arrangement we see that each driving arm, RP and RQ' , has a period of almost complete rest and two periods of rapid motion. The rapid motion period effects a rapid opening of the port to its full width and then a rapid closing, the period of comparative rest occurring when the port is closed by valve. This arrangement of bent levers does excellent service but modifies greatly the "harmonic motion" of the driving point of the link and requires other diagrams than the valve-circle for the exact representation of the valve travel. We will return to this matter later on.

We will first show that the travel of a point on the link EGL is approximately like the "harmonic motion" communicated by an eccentric that has an infinitely long eccentric rod. Let A represent the distance from center of eccentric B to center of trunnion E , the latter joining sustaining arm FE and link EGL on center line of slot of link; let ρ represent the eccentricity CB , u the distance of point G above dead center line of eccentric, ω the crank (or eccentric) angle and ξ the distance of eccentric center B from perpendicular through C to eccentric's dead point line (which is approximately along CEO' , the crank and eccentric differing in position by a small angle of $2^\circ 12'$ that is equal to angle included between their respective dead point lines). Then,

for the case in which $\frac{A}{\rho} \geq 4$, we will show that approximately:

$$\xi = \rho \cos \omega + \frac{u}{A} \rho \sin \omega \quad (90)$$

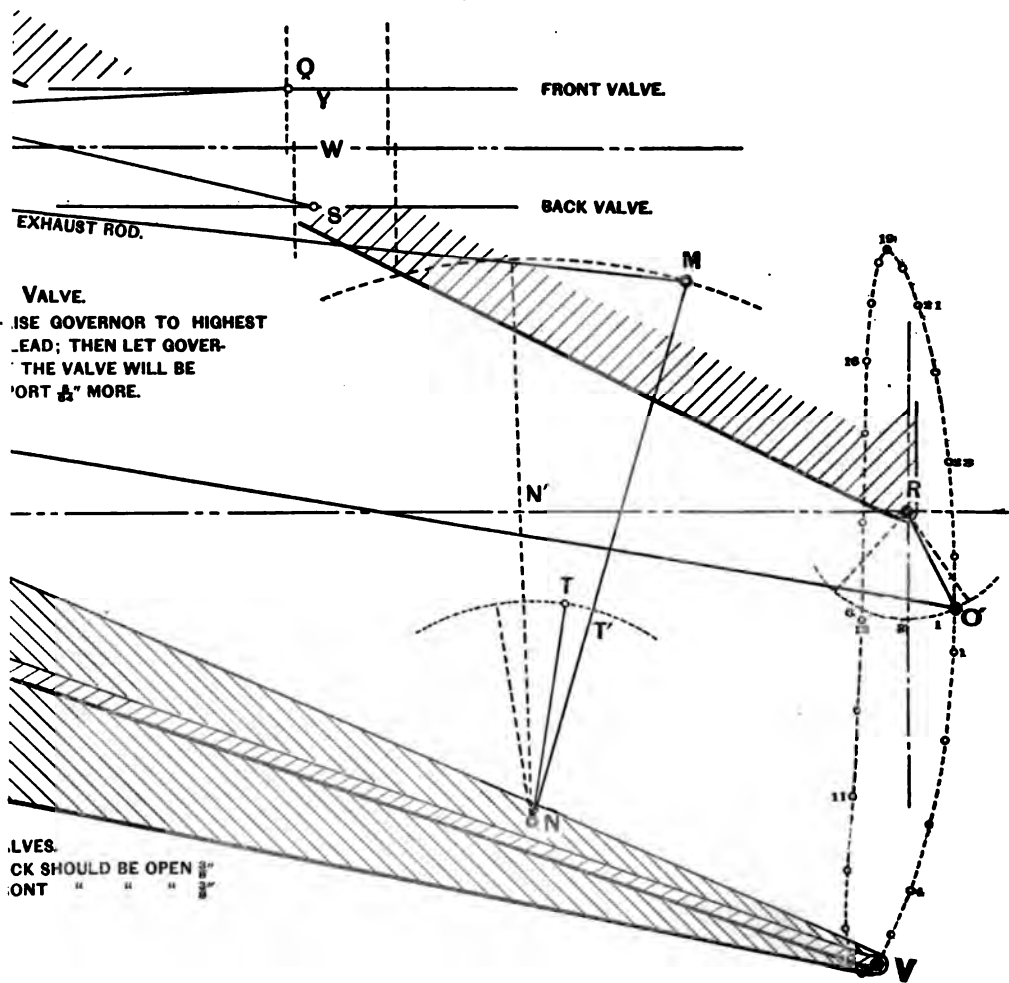
To prove this, we consider the total movement of link to be composed of two parts, of a *horizontal* motion in which the link moves parallel to itself, the character and extent of this motion depending on the *horizontal* throw of the eccentric, and of a *rocking* motion about the trunnion E (or vibrating fulcrum) which is due to the vertical throw of the eccentric. Here the term $\rho \cos \omega$ is the horizontal throw common to all points of the link and the term $\frac{u}{A} \rho \sin \omega$ is the vertical throw $\rho \sin \omega \times$ the ratio $\frac{u}{A}$ of the arms of the bent lever constituting the link and rotating about the fulcrum or trunnion E . The formula for ξ thus established is the polar equation of the valve circle and the coordinates ρ and $\frac{u}{A} \rho$ of the vertex of its diameter show that this vertex lies on a rectilinear locus perpendicular to the dead point line of the eccentric and at a distance ρ from the center of the shaft. Such a valve circle is drawn in Fig. 83 for full gear (*i. e.*, for block G about 6" from trunnion), and the corresponding equivalent, (virtual or resultant) eccentric can be found in the usual way, Fig. 60, from the diameter of this valve circle.*

To show that the valve circle thus found represents with considerable accuracy the travel of a point on the link, we

* As the horizontal component motion due to the rocking, is motion relative to trunnion E , we can also combine the parallel and rocking components by means of the parallelogram of velocities. We need only regard the rocking component of the motion as directly and independently produced by a separate eccentric, one that would *follow* the actual eccentric by 90° and have a length $\frac{u}{A} \rho$. As these two eccentrics are both at right angles to, and proportional to, the velocities they impart we can construct the parallelogram of velocities on the eccentrics as sides and the diagonal will give the length and location of a resultant or virtual eccentric capable of imparting directly the whole horizontal motion to the point of the link. The parallelogram and triangle of eccentricities constructed in Fig. 83 and Fig. 84, respectively, are illustrations of this method. The valve-circle can then be found from the virtual eccentric as in Fig. 60.

have in Fig. 83 compared such a valve circle with a polar curve OMN or $OM'N'$ representing very exactly the travel of the rocker-pin O' . (Figs. 79 and 82). To simplify matters we have assumed pin O' to travel on a straight line that bisected the rise of its arc-path and passed through center C (Fig. 82) of shaft. In Fig. 83 we have called this straight line CO' (of Fig. 82) the dead center line of eccentric and constructed the valve circle according to formula 90 found above. The dotted, polar, curves OMN and $OM'N'$ (Fig. 83) were found by laying off on the crank positions the exact distance of pin O' from its center of motion, supposing O' to travel on aforesaid dead point line CO' . We see that these curves agree very well with the valve circles and that the latter may unhesitatingly be used to represent the travel of lower rocker-pin O' parallel to dead point line $O'C$. Inspection of the setting of the upper rocker pins Q' and P shows at once that the horizontal components of their motion could not be represented by valve circles, and that to represent the valve travel accurately we must construct exact polar curves for both the front and the back valve.

A fairer comparison of the actual travel of a point on the link with that given by the formula or valve-circle is furnished by Fig. 84, which represents by the dotted, polar, curves the actual travel, from its center of motion, of the point T (Fig. 82) driving the exhaust valve. It is evident that the average direction of the point M of the exhaust rocker NM is parallel to the average direction of the path of the driving point L on link. We may therefore assume that the actual travel of M is almost exactly like that of L . Now the travel of T is 0.39 of that of M ; it is represented by the dotted, polar, curves of Fig. 84, which were obtained by directly measuring the distances of point T from its center of motion and then laying off the distances on the corresponding positions of the crank. The valve-circle representing, approximately, the travel of point L is drawn on diagram on diameter OD_v' and reducing it 0.39 we get the valve-circles



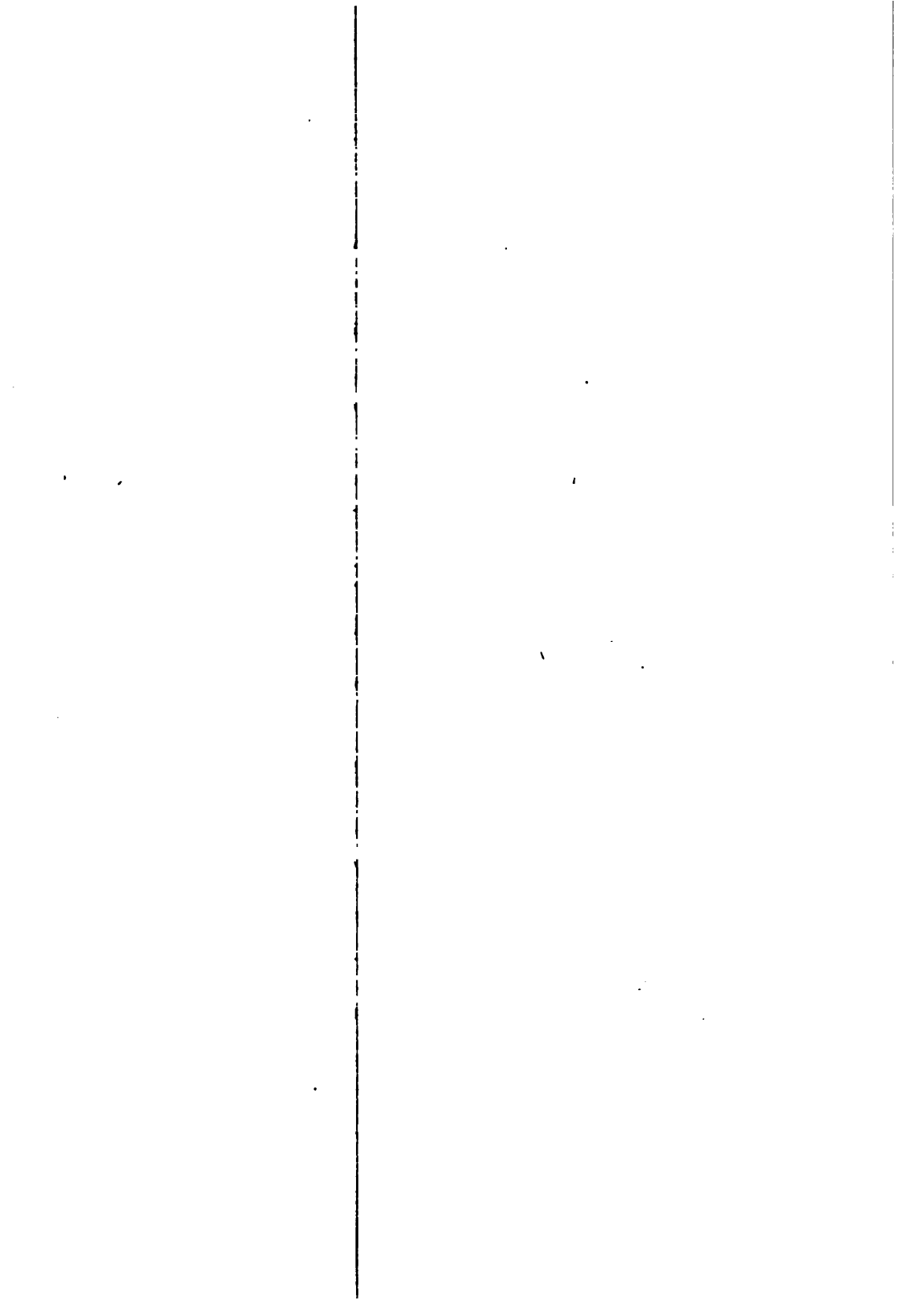
OD_e'' representing, approximately, the travel of driving point T . Though the deviation of the two sets of curves is now marked, nevertheless, the deviations occur where they will but slightly influence the beginning of compression or release, which are found as of old at the intersections of the polar curves (or circles) with the lap circles.

We have already called attention to the fact that the setting of the driving arms PR and $Q'R$ causes them to communicate to the valves a horizontal motion that is decidedly different from the "harmonic motion" assumed for eccentrics. If we wish to represent the valve's motion accurately by polar curves whose vectors are the crank positions, we must determine accurately the distance of the valve from its center of motion for a complete series of crank positions.

Inspection of the skeleton in Fig. 82 shows that it is composed of two quadric chains $CBEF$ and $HIRO'$ connected by the straight link GV (which in the actual mechanism is represented by the link-block G). For the assumed point of suspension I the point G of second quadric chain describes the path $G, 1, 3, 6$, while the point O' of lower rocker arm describes the corresponding path $O', 1, 3, 6$. In the first quadric chain $CBEF$ the point L occupies points 1, 8, 13, 23, etc., of its path, while eccentric center B occupies points 1, 8, 13, 23, etc., of its circular path. The center V of the slot GE is the point on which rod GV turns and this point is on the line BE of the first quadric chain; the piece BE in the skeleton is supposed to be extended so as to support this point V . The path V is given and the numbers correspond to the crank or eccentric positions. We can easily find the position of either valve for any crank position if we know the corresponding position of O' . To get O' for any eccentric (or crank) position, we suppose V placed at that point of its path which has the same number as the crank position under consideration; with this point V as a center strike off an arc cutting the path of G ; then with this intersection of the arc and the path as

a center and the steam rod GO' as a radius strike off another arc; it will cut the circular path of O' at the point O' desired.

The position of each valve can be represented by its point Q or S and is estimated from a center of motion that is invariable in position for all grades of expansion, *i. e.*, for all positions of the block G in its slot. To find this center of motion we must first find two peculiar eccentric positions *near* the dead point line of the eccentric. To get these positions we erect at C (Fig. 82) the perpendicular $CF_1 = EF$ to the dead point line of eccentric. Then with F_1 as a center and F_1C as a radius describe an arc tangent to this dead point line; the chords $C1$ and $C13$ of this arc will be the peculiar eccentric positions needed. Now find two positions of both Q and S corresponding to these two peculiar eccentric locations and then find a point half way between them; this point will be the desired center of motion from which the travel is to be estimated, and will be found to be the same whatever position is assumed for the point G in its slot (that is, it will be the same whatever fixed position is chosen for the point of suspension I of the hanger IH .) The center of motion can also be found, and more simply still, in the following way. The steam rod is $\frac{3}{16}$ " shorter than radius of link slot; near the two intersections of path of center V (Fig. 82) with arc of O' lay off $\frac{3}{16}$ " from arc O' towards path of V , in the direction of middle radius of the link slot, till a place is found (on each half of V) where arc O' and path of V are just $\frac{3}{16}$ " apart. When center of steam rod occupies either of these two positions of O' the block may be moved from one end to the other of link slot without moving rocker pin O' or either of the two steam valves. Hence these two points on arc O' will give exactly the two valve positions that are equidistant from the center of motion; in like manner the two points on path V thus found will give the corresponding eccentric positions and these will be found to be near the peculiar ones described above. The distance of each valve from its center of motion is now laid off on the correspond-



GRAMS FOR BOTH STEAM VALVES—

LAP = $\frac{25}{32}$

HALF GEAR

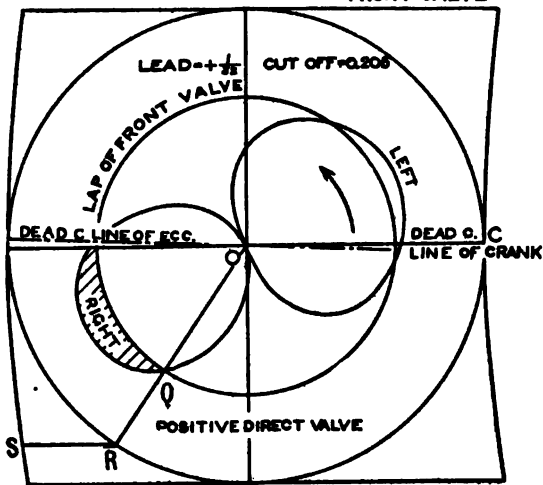
FRONT VALVE

2-8 CRANKS

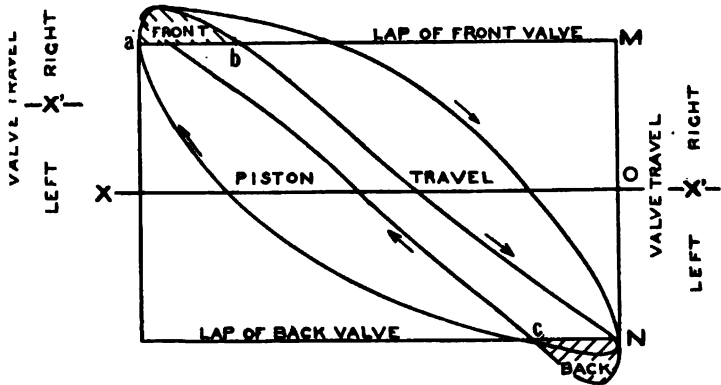
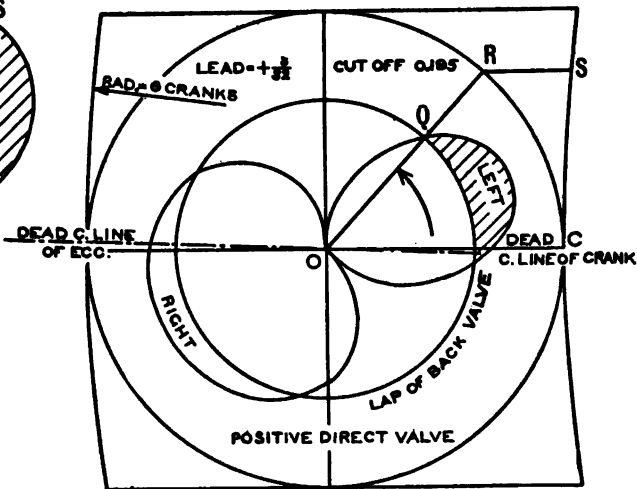
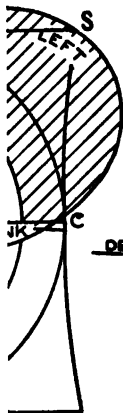
AD C. LINE OF CRANK

AD = $-\frac{1}{4}$
IT OFF = 0.511

$\frac{L}{R} = 6$



BACK VALVE



ing crank positions and the polar curves shown in Figs. 86 and 87 for full gear (G is then 6" from trunnion E) obtained, also the polar curves shown in Figs. 89 and 90 for half-gear (block G is 3" from trunnion E). To ascertain the lap from the data given in Figs. 79-82, we must find positions of point $\left\{ \begin{smallmatrix} Q \\ S \end{smallmatrix} \right\}$ when engine is on $\left\{ \begin{smallmatrix} \text{forward} \\ \text{back} \end{smallmatrix} \right\}$ center and block G at trunnion E ; as the edge of the $\left\{ \begin{smallmatrix} \text{front} \\ \text{back} \end{smallmatrix} \right\}$ steam port is then $\frac{1}{16}$ " $\left\{ \begin{smallmatrix} \text{left} \\ \text{right} \end{smallmatrix} \right\}$ of edge of $\left\{ \begin{smallmatrix} \text{front} \\ \text{back} \end{smallmatrix} \right\}$ steam valve, we lay off $\frac{1}{16}$ " to the $\left\{ \begin{smallmatrix} \text{left} \\ \text{right} \end{smallmatrix} \right\}$ of $\left\{ \begin{smallmatrix} Q \\ S \end{smallmatrix} \right\}$ to get its position when valve edge is exactly at port edge. Now place $\left\{ \begin{smallmatrix} Q \\ S \end{smallmatrix} \right\}$ at its own center of motion and measure its distance from its former position when valve and port edges coincided; the distance will be the steam lap desired. Laying off the same steam lap = $\frac{3}{16}$ for both front and back valve, and measuring in each case the distance RS (Figs. 86-90) and dividing it by the stroke = $(2 \times \overline{CO})$, we get the cut-off effected at each end and at each grade of expansion. The shaded areas give in all cases the linear port openings and should be multiplied by 4 for the total openings. The numerical values of the distribution are inscribed on the diagrams and show that it is a practically perfect valve gear in this respect.

The actual valve travel found above can also be laid off as ordinates, on the piston stroke as a base, and then we get another series of valve diagrams of an oval shape, that are known as "motion" curves. The exact polar and oval diagrams give of course identical results. Figs. 85, 88 and 91 respectively represent the travel of the exhaust valve, that of the front and back steam valves for full gear and that of the front and back steam valves for half gear. In Fig. 85 cushioning in crank end is measured by $\overline{aX} \div \overline{OX}$, release in head end by $\overline{bX} \div \overline{OX}$; similarly cushioning in head end is given by $\overline{dX} \div \overline{OX}$ and re-

lease in crank end by $\overline{cm} \div \overline{OX}$. In Figs. 83 and 91, $\overline{ab} \div \overline{OA}$ measures the cut-off in crank end and $\overline{Ac} \div \overline{OX}$ the cut-off in head end. The polar diagrams for the steam valve best show the variation of lead. The speed of this size of Porter-Allen engine is 230 revolutions per minute.

DOUBLE-VALVE GEARS.*

The characteristic of a double-valve gear is that while the first of its two valves slides on a fixed seat, the second slides on the first, the motion of the second valve relatively to the first being affected by the absolute motion possessed by each of the two valves.

In single-valve gears with variable elements, and in link-motions, we simplify our problem by finding the equivalent or virtual eccentric and then proceed as if the valve were directly moved, to and fro upon its fixed seat, by this virtual eccentric. In double-valve gears a like simplification is introduced, in fact it is the finding of this virtual eccentric which usually constitutes

* This term, double-valve, has occasionally been applied to a valve consisting of two rigidly connected halves and also to two separate and independent valves each sliding on a fixed seat. The gear here called double-valve gear has sometimes been defined as gear with plain slide valve and independent cut-off; it has also been described as composed of a plain slide valve and a riding valve. But these are long expressions and are also not free from ambiguity.

We wish to distinguish between gears requiring consideration of only absolute motion and those in which relative motion must be considered. Gears possessing but a single valve evidently belong to the former class and the term, single valve, is sufficiently suggestive of absolute motion. As relative motion implies at least two pieces we feel justified in using the term double-valve to suggest this motion. A further justification is supplied by the fact that the principal functions, of opening or closing a steam passage, must be doubly performed (though not simultaneously) when one valve slides on the back of another. We must confess however that we are not satisfied with the terms we have chosen and hope that something else as simple but more exact may be found.

the main part of the problem. Such an eccentric drives the expansion valve, over a perfectly stationary main valve, with an absolute motion that is the same as the motion of the expansion valve *relatively* to the main valve when both these valves have their actual motion. In this way the new and complex problem of the relative motion of two valves, is reduced to the old and simple problem of a single-valve sliding on a fixed seat and driven by one eccentric.

Now as to the method of procedure. When the main and expansion eccentrics are both given, the virtual eccentric can easily be found by means of a proposition for which Dr. Zeuner suggests the name of "Parallelogram of Eccentricities."

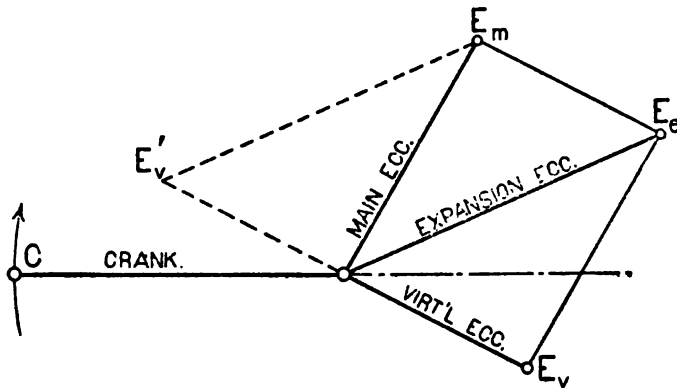


Fig. 92.

It may be stated as follows: *The main, virtual and expansion eccentrics constitute, respectively, the two adjacent sides and diagonal of a parallelogram.*

In Fig. 92, OE_m , OE_v and OE_e represent these eccentrics and $OE_m E_v E_e$ the parallelogram which they form. To avoid confusion hereafter it should be specially noted that the *expansion eccentric* is always the *diagonal* of this parallelogram, provided the virtual eccentric OE_v effects the motion of the expansion valve relatively to main valve. But when it is a question of

relative motion of main valve to expansion valve the virtual eccentric must be OE_v' , just opposite to OE_v , and OE_m must then be the diagonal of parallelogram OE_v, OE_m, OE_v' . In all our work we shall take the *expansion eccentric* as the *diagonal* of the parallelogram.

In *existing* valve gears the main and expansion eccentrics are completely known and the examination of the steam distribution effected by their valves is easily made by means of the main and virtual valve-circles, which are deduced in the usual way from the known main and virtual eccentrics.

In *designing* valve gears, however, the expansion eccentric is the unknown quantity whose variations must satisfy the prescribed conditions as to range of expansion, time of admission, rapidity of cut-off, etc. To find it we must first find the main and virtual eccentrics and with these two as adjacent sides construct a parallelogram whose diagonal will be the desired expansion eccentric.

The main eccentric can be found in the usual way from its valve-circle and this in turn from the given conditions as to lead, port opening, maximum cut-off, release or compression. The virtual eccentric is found in a similar way, its circle being first obtained from such given conditions as, type of expansion valve, cut-off and time of reopening.

Having thus indicated the principal steps in designing a double-valve gear, we will return to the beginning and show that, when there are two valves and both are in motion, one riding on the other, then a fixed circle can be found which will cut from the crank, or its prolongation, chords whose lengths represent the distances between a point on the expansion valve and a corresponding point on the main valve, the correspondence being simply that the two points are together when the expansion valve occupies its middle position on its seat on the main valve. The distance between two such points at any crank position is evidently equal to the (algebraic) difference of their distances

from any common, fixed, reference line. Neglecting the angularity of the eccentric rods, this difference is equal to the difference in the distances $E_e S_e$ and $E_m S_m$, Fig. 93, of the corresponding eccentric centers E_e and E_m from their common reference line OP , which is perpendicular to the dead point line NOZ of the eccentrics. Our problem is now reduced to finding a

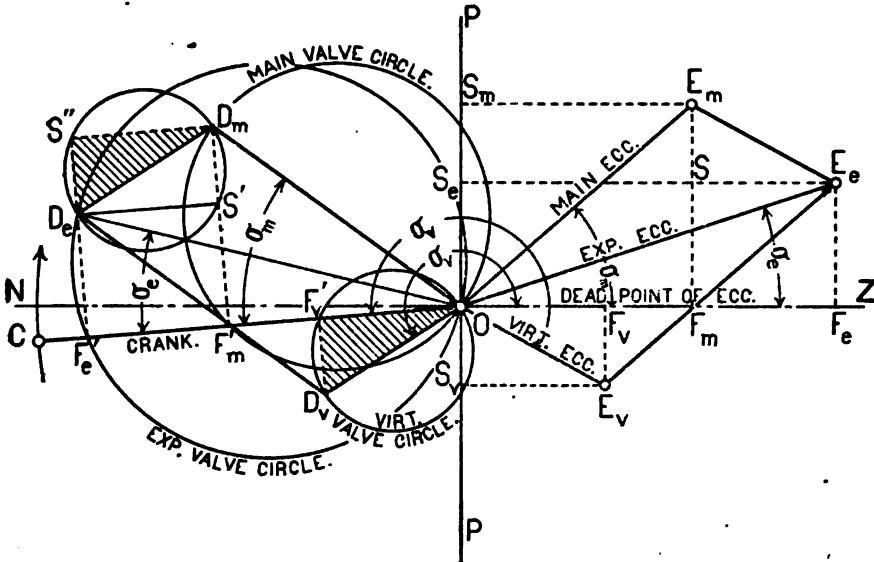


Fig. 93.

circle that will cut from any crank position OC a chord equal to the corresponding difference $E_e S_e - E_m S_m = E_e S$ of the distances of the centers E_e and E_m from the middle position OP of the eccentrics.

Finding the main valve-circle OD_m from its eccentric OE_m as in Fig. 60, p. 137, we get the chord $OF_m' = OF_m = E_m S_m$ as the distance of E_m from OP . The distance $E_e S_e$ of E_e from OP can be found in the same way by constructing the expansion valve-circle OD_e . To do this we measure angle σ_e from the right portion of the eccentric's dead-point line OZ , in a direction

opposite to the rotation, and lay off this angle σ_e from the crank in the direction of rotation; $OD_e = OE_e$ will be the diameter of the expansion valve-circle and $OF'_e = OF_e = E_e S_e$. Then because $D_m S''$ is parallel to crank position OC , we have $D_m S'' = F'_e F'_m = F_e F_m = E_e S$. As $D_m F'_m$ is perpendicular to OC , the angle $D_m S'' D_e$ is a right angle. But the hypotenuse $D_e D_m$ is constant in position and magnitude for all crank positions; the locus of S'' is therefore a circle. If through D_m we draw a parallel to the crank position OC at any instant, this circle will cut from this parallel a distance $D_m S''$ which is equal to the value of $E_e S$ at this instant. By moving this circle parallel to itself so that the point D_m of this circle $D_m S' D_e$ shall shift its position to shaft center O , the circle will take up the new position $OD_v F'_v$ and cut from the crank itself (or its prolongation) the distance $OF'_v = D_m S'' = E_e S = E_e S_e - E_m S_m$. In its new position it may be called the *virtual valve-circle* for it possesses all the properties of a valve-circle relatively to its eccentric. For instance, the virtual eccentric OE_v is found from the virtual valve-circle OD_v by laying of the angle σ_v in the same way as the angles σ_m and σ_e . Moreover, when the $\left\{ \begin{array}{c} \text{actual} \\ \text{prolonged} \end{array} \right\}$ crank cuts the virtual valve-circle, the virtual eccentric will be to the $\left\{ \begin{array}{c} \text{right} \\ \text{left} \end{array} \right\}$ of its middle position OP . As regards the periods during which the riding-valve keeps the ports in the main valve opened or closed, the virtual valve-circle, like the two others, is subject to the conditions of valve-type, etc., laid down in Table XIV. The valve driven by the virtual eccentric is of course of the same type as the expansion valve, the *relative* motion of the latter corresponding exactly to the *absolute* motion of the former.

Before applying these results to the common high-speed engines, there is still another proposition enunciated by Dr. Zeuner which is of the greatest service in the solution of valve-

gear problems. But before stating what it is, we will point out that with double-valves the expansion can be varied :

- I. By varying the angle of advance of the expansion eccentric.
- II. By varying the throw of the expansion eccentric.
- III. By varying both angle and throw of the expansion eccentric.
- IV. By varying the lap of the expansion eccentric.

In the last case neither throw nor angle of advance of expansion eccentric is varied. The most general case is therefore III ;

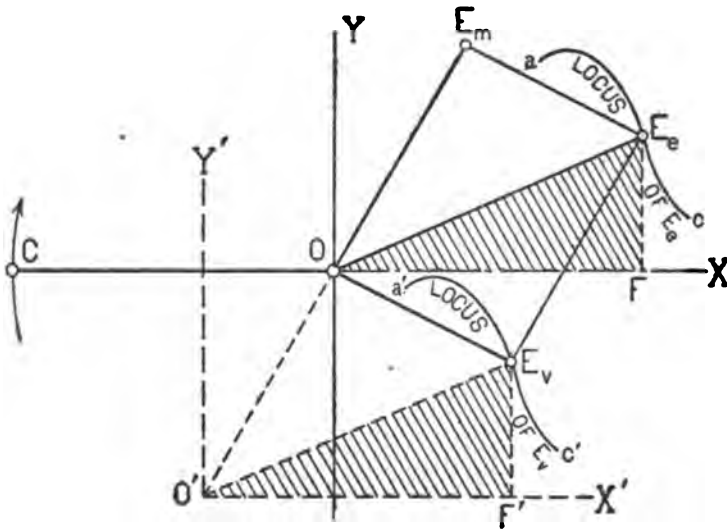


Fig. 94.

it is represented in Fig. 94, where the center E_e traverses any arbitrary locus ac . Its reference line is taken to be the crank at left dead point, but any other crank position might have been chosen and the locus ac would occupy a correspondingly different position in the plane of the paper. It is simplest to consider the locus ac as rigidly attached to the crank, moving with it, and preserving always the same relative position towards it.

Let OX and OY be coordinate axes rigidly attached to the crank and moving with it, then will OF and FE_e represent the

coordinates of any point E_e of the locus ac . Prolong main eccentric $E_m O$ to O' , making $OO' = OE_m$, and draw a parallel set of coordinate axes $O'X'$, $O'Y'$, also rigidly attached to crank. Then the proposition of Dr. Zeuner, last referred to, consists in the statement that, *The locus $a'c'$ of the center E_v of the virtual eccentric OE_v is a curve exactly equal and parallel to the locus ac and the position of $a'c'$ relative to the origin O' is exactly like that of ac relative to origin O .*

To prove this it is only necessary to show that the coordinates $O'F'$, $F'E_v$ are respectively equal and parallel to coordinates OF and FE_e . By construction OO' is equal and parallel to $E_e E_v$, which makes $OO'E_v E_e$ a parallelogram, hence OE_e is equal and parallel to $O'E_v$ and this makes the two sets of coordinates respectively equal and parallel. Since in designing we start with the valve-circles rather than with the eccentrics it will be convenient to represent this proposition as applied to the diameters of the valve circles.

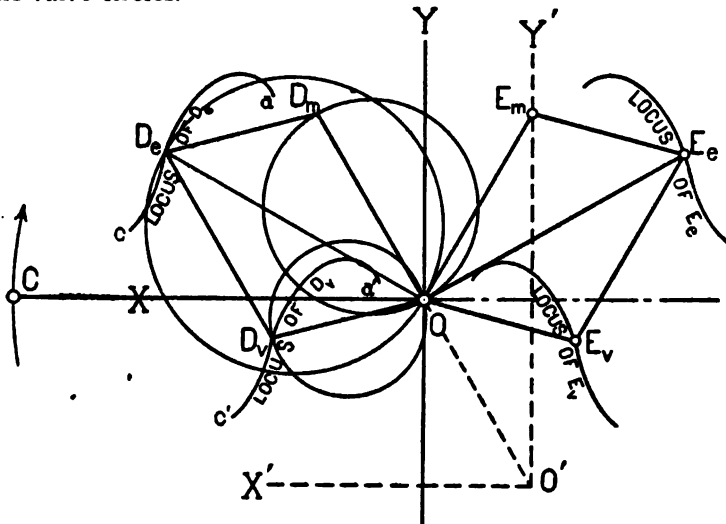


Fig. 95.

In Fig. 95 the curve ac is the locus of the vertex D_e of the valve circle diameter OD_e . This curve ac is equal to the locus of

E_e and is similarly located. The curve $a'c'$ is the locus of the vertex D_v of the diameter OD_v of the virtual valve-circle, and is equal and parallel to the locus of E_v . The locus $a'c'$ has the same position relatively to the origin O' that ac has relatively to the origin O . These valve-circle-loci, unlike those in Fig. 94, have the great advantage of preserving their places in the plane of the paper during the rotation of the crank.

The subscripts e, m, v employed in these diagram relate, respectively, to expansion, main and virtual eccentrics and their valve circles. The symbols e_o, r_o and δ_o respectively refer to the lap of the expansion valve, the eccentricity and angle of advance of the expansion eccentric. When the subscripts 1 and 2 are annexed to the subscript o the values at minimum and maximum grade of expansion are meant. The other symbols have the meaning already assigned them in the discussion of single-valve gears.

The loci of the center of expansion eccentric vary greatly in form according as the valve-gear elements, angle of advance, throw and lap are varied. In Figs. 100 and 101, representing the expansion valve diagrams of the Cummer engine, the locus of E_e is circle concentric to shaft center O and has r_o as a radius. In Fig. 106, the expansion valve diagrams for a Meyer valve gear, this locus reduces to a point, for the setting and throw of the expansion eccentric does not vary with the grades of expansion but only with the lap. The inscription, "locus of D_v ," on this diagram is to be understood only as a geometrical construction that is helpful in solving the problem. In Fig. 109, the expansion valve diagram of the Rigg engine, the locus E_e is OE_m, E_m , a straight line passing through O . In Fig. 112, the locus of E_e is a circle that is not concentric to O . In Figs. 115, 116 the locus E_e is a straight line that does not pass through O and in Figs. 118, 119 locus of E_e is approximately a circle whose circumference passes through shaft-center O .

In designing a double-valve gear, the steps are: (a) to find the

main eccentric suitable for release, compression, and beginning of admission, (*b*) to find the virtual valve circle that will give the desired cut-off, reopening, etc., for the assumed type of expansion valve, (*c*) to find the corresponding virtual eccentric, and (*d*) to find the desired expansion eccentric by combining the main and virtual eccentrics.

When very great accuracy in the determination of valve travel is desired, we must substitute for the virtual valve-circle a pair of exactly drawn curves whose vectors represent at once the crank's position and the valve's travel from its center of motion (see definition of this center, p. 151). To construct these polar curves we must therefore find this valve travel. Accordingly we should, in Fig. 93, replace the middle position or reference line *POP* of the main eccentric by a reference *arc* struck from the center of motion with eccentric rod as radius. The distance of OE_m from this arc will be $E_m S'_m$. In like manner we must replace the reference line *POP* of the expansion eccentric by the reference arc (of the valve) struck from the corresponding center of motion and the eccentric rod as radius. The distance of E_e from this arc will be $E_e S'_e$ and the true relative position or travel of the expansion valve will be $E_e S'_e - E_m S'_m$ and this difference laid off on the corresponding crank position will give one point on the desired polar curve.

The polar curve obtained by laying off on the crank the distance $E_v S'_v$ (of the center E_v of the virtual eccentric from the common arc of reference) is in general not a closer approximation than the valve-circle itself.

When there is a reversing lever between one valve and its eccentric, the actual distance of each from the arc of reference must still be taken, remembering however to take the $\left\{ \begin{array}{c} \text{difference} \\ \text{sum} \end{array} \right\}$ of $E_e S'_e$ and $E_m S'_m$ when the *valves* are on the $\left\{ \begin{array}{c} \text{same side} \\ \text{opposite sides} \end{array} \right\}$ of their center of motion.

E. EXPANSION

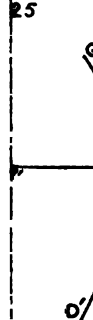
DIRECT VALVE.



**THE DIRECT VALVE
CUT-OFF = 0.5**



DIRECT VALVE.



REVERSING ARMS BETWEEN ECCENTRIC AND VALVE

In the great majority of cases the valve-circles will be sufficiently accurate; in the few remaining cases it will rarely be necessary to construct the whole polar curve; a few points of the curve (for the crank positions of the most practical interest) will usually suffice.

EXAMPLES OF DOUBLE-VALVE DIAGRAMS, FIGS. 99-119.

In these mechanisms the expansion valve moves on the main valve and controls the ports or steam passages in the latter. Since both valves move their relative motion depends upon the separate, absolute, motions and is one of some complexity, but it can be simplified by considering the main valve stationary and supposing the expansion valve moved by a new special eccentric that imparts to it an absolute motion equal to the relative motion which it possesses when both valves move under the influence of their own, actual, eccentrics. This eccentric has been called the virtual eccentric and for a particular grade of expansion has all the simplicity and properties of the ordinary fixed eccentric. Its valve-circle is found from the virtual eccentric in the usual way and gives the same sort of information concerning the crank positions at which the port in the main valve opens and closes.

In this set of diagrams the port openings have not been indicated by sectioned areas. With negative valves the port opening is measured *inward*, from negative lap circle to valve-circle, this intercept lying wholly or partly *outside* of the valve-circle.

When there is a reversing arm between an eccentric and its valve, we assume in diagramming the same type of valve but an eccentric capable of giving the valve motion directly, without such an intermediate lever. At the end of the procedure this is allowed for by placing the resultant expansion eccentric 180° from that given by diagram.

As this series of figures (99-119) is intended to illustrate the double-valve system of varying the expansion, we have neglected the exhaust side of the distribution in our main valve diagrams. But this may easily be supplied from what has gone before.

BUCKEYE ENGINE, 13x24, FIGS. 96, 97, 98.

We have already touched upon the kinematic character of these valves in the foot-note, p. 154.

Generally when there are two valves there are two eccentrics and these usually give a resultant or virtual eccentric, but not always. There is one exception in the Buckeye Engine. In this case the mechanism between expansion eccentric and expansion valve is such that the rod of this expansion valve receives a motion nearly equal to that of the main valve thus making the relative motion of the expansion valve on the main valve dependent simply on the motion of the expansion eccentric. The latter therefore is at once expansion and virtual eccentric. In this case of double-valves and double eccentrics we need speak only of the *expansion* eccentric and valve-circle.

Fig. 96. The main valve in this case covers the port in its middle position and cuts off with its inner edges which brings it into the Positive Indirect class of valves. The location of the corresponding valve-circle OFD_m , Fig. 96, can easily be chosen by the help of Fig. 62 and the remarks made on pp. 142, 143. The eccentric can now be found as in Fig. 60. It is evident from Table XIV that left cylinder is open for steam admission between crank positions 3 and 4 and closed during crank's motion through positions 4 — 1 — 2 — 3, closing at 4 and reopening at 3.

Fig. 97. For the sake of economy of construction we will take $r_o = r$ and describe a circle with O as center and r as a radius. To find the valve-circle for minimum cut-off, when negative lap e_o is given, we erect at H a perpendicular HD_r to dead

center line OH of crank. Then will OD_{e_1} be the diameter of the expansion valve-circle for minimum cut-off. Inspection of Table XIV and figure 97 shows that this negative direct valve keeps its left port in the main valve closed from crank position 3 to 4 and open during 4 — 1 — 2 — 3, the reopening taking place at 4 after main valve has closed the left cylinder port. In getting the position of the expansion eccentric it should be noticed that in this particular problem the dead center lines of eccentric and crank do not coincide differing by about $8\frac{1}{2}^\circ$. After the manner of Fig. 60, we find OE_{e_1} to be the position of expansion eccentric at minimum cut-off, when crank is at left dead point. As there is a reversing lever between eccentric and valve, we must reverse OE_{e_1} to get position OE'_{e_1} of expansion eccentric in the actual case.

Fig. 98. In order that the expansion may have the widest possible range the expansion valve at maximum cut-off should close its port at the same instant that the main valve closes the corresponding cylinder port. To realize this place the crank at position belonging to main-valve's cut-off and then find, according to Fig. 62, the location of the proper valve-circle. As the lap e_o does not change the perpendicular ID_{e_2} to the crank position CO_1 will give OD_{e_2} as the diameter of the valve-circle desired. Laying off the proper angles as in Fig. 60, we get OE_{e_2} for the position of expansion eccentric at maximum cut-off, when crank is at cut-off position and no reversing lever exists between eccentric and valve. As there is such a reversing lever, we must change by 180° the position of OE_{e_2} and thus get OE'_{e_2} , the position of the actual expansion eccentric when crank is at position of maximum cut-off. For this same position CO of the crank the main eccentric is at OE_m and the expansion eccentric's position corresponding to minimum cut-off is at OE'_{e_1} . The angles of advance δ_o for the extreme cut-offs are easily obtained provided the definition of this angle, given on p. 150, is borne in mind. The angle E'_1, OE'_{e_2} measures the variation of angular position of

expansion eccentric for the whole range of cut-off. Table XIV again shows that the left port in main valve is open during the crank's motion 4 — 1 — 2 — 3, and there will therefore be a large lead by the expansion valve at the beginning of the forward stroke. The cut-off given on this set of diagrams are approximate only. They can be obtained more accurately from the accompanying piston travel diagram in which $\frac{L}{R} = 6$. In this connection see method of finding cut-off in Fig. 63.

CUMMER ENGINE, 11x20, FIGS. 99, 100 AND 101.

The functions of the steam distribution are divided among three pairs of valves, three valves for each end of the cylinder. There is a so-called main valve which however only regulates the beginning of the admission of steam to the cylinder, though it also has some influence on the relative velocity with which the cut-off valve opens and closes its ports in the main valve. Next there is the cut-off or expansion valve which as its name implies regulates only the grade of expansion. Finally there is the exhaust valve which regulates the beginning of release and compression for each end of cylinder making them both constant for all grades of expansion. In each of these three pairs of valves, the two halves are rigidly connected. There are at each end *three* steam ports and *four* exhaust ports, all valves being of the gridiron variety. None of these flat valves are balanced, because the valve stroke is small and therefore the work of friction so small that there would be little gained by balancing. The port openings are large for both admission and exhaust.

In this engine the expansion is varied by changing the angular position of the eccentric, as in case of the Buckeye Engine. In the case of the Cummer Engine however, the expansion and virtual eccentric are not identical, the latter being a side of the parallelogram of eccentricities constructed on main and expan-

sion eccentrics, as in the full lines of Fig. 92. Here the complex motion of two valves is reduced to one valve driven by an equivalent eccentric called the virtual eccentric. When this is known the expansion eccentric is known and the problem is solved. The principal steps of the solution are: (a) To find the locus of the vertex of the diameter of the virtual valve-circle. (b) To construct these valve-circles for the extreme cut-offs. (c) To find the eccentrics from the diameters of the valve-circles.

Fig. 99. The construction of the main valve diagram presents nothing new, We may therefore omit a description of it. We simply call attention to the fact that with this positive, direct, main, valve the left cylinder port is open from crank position 1 to position 2 and closed more than half the revolution, namely 2 — 3 — 4 — 1, (see Table XIV). This and the expansion diagrams are drawn to twice the usual scale because of the smallness of the elements of the problem. The lead v is only $\frac{1}{32}''$ instead of $\frac{1}{16}''$ as given in diagram. The width of port is $\frac{3}{8}''$.

The exhaust valve's diagram has not been given, though it deserves attention because there is a reversing lever between it and the eccentric of the main valve. This eccentric drives both the exhaust and the main valve, but the latter is driven directly. These two valves are both of the positive direct type, but move always in exactly opposite directions. If the exhaust valve kept its present motion but were driven directly its eccentric would have to be 180° from its present position and its valve-circle would be drawn 180° from the equal one shown in Fig. 99. But this figure, as it stands, may be used for the exhaust valve provided we (draw the inside lap and) remember that then the $\left\{ \begin{array}{l} \text{actual} \\ \text{prolonged} \end{array} \right\}$ crank cuts the present valve-circle when exhaust valve is to the $\left\{ \begin{array}{l} \text{left} \\ \text{right} \end{array} \right\}$ of its middle position. The lap on the exhaust side may easily be ascertained by assuming that cushioning takes place during the last fifth of the stroke; the beginning of the release can be determined from the inside lap thus found.

Fig. 100. Given e_o and r_o . As the eccentricity r_o of the expansion eccentric does not change, the locus of the center of the latter will be a circle FLM described about O with radius r_o . The vertex of the diameter of the expansion valve-circle has the self-same locus FLM about O as a center. Consequently according to principle enunciated in connection with Fig. 95 the locus of the vertex D_v of the diameter of the virtual valve-circle must be a circumference NQP struck with r_o as a radius and O' as a center. As the negative lap is given, we have only to erect the perpendicular HD_{v_1} to the crank position COH at minimum cut-off to get a second locus of D_{v_1} , and thus get the diameter OD_{v_1} of the desired valve-circle. We see that the reopening of the expansion valve takes place just a little before the crank reaches the dead point. By laying off the angles as in Fig. 60, we find OE_{v_1} as position of virtual eccentric for $\frac{1}{10}$ cut-off and then, by means of the parallelogram, OE_{r_1} as the position of the corresponding expansion eccentric when crank is at dead point.

Fig. 101. Given e_o , r_o and the extremes of cut-off. In like manner we find OE_{r_2} is the position of expansion eccentric at the maximum cut-off. If we transfer the eccentric position OE_{r_1} in Fig. 100, for minimum cut-off and crank at dead point, to the present figure, the angle $E_{r_1}OE_{r_2}$ will measure the total change of angular position of the expansion eccentric for this range of cut-off.

Two limits may be placed upon the time of reopening of the expansion valve. The first limit is set by the condition that the reopening of its port at either end by the expansion valve must surely occur *after* the main valve has closed the steam port at that end; otherwise there will be two admissions of steam during the same stroke, and the benefit of expansion will be destroyed. The second limit is prescribed by the condition that, for the same cylinder end, the expansion valve must open port as soon as, or sooner than, the main valve. The two crank positions at which the positive main valve cuts off and admits steam at the

same port, are always more than 180° apart. Hence, with a negative direct expansion valve, we have only to guard against a premature opening and therefore need to consider only the first of the two limits mentioned above.

Let us assume new data, namely, that r_o is given, also the extremes of cut-off ϵ_{oi} and ϵ_{oo} and the crank position at which main valve cuts off; required a suitable lap ϵ_o .

Now the diameter of a valve-circle lies half way between the two vectors that contain its intersections with the lap circle. If for minimum cut-off, we bisect the angle between the crank at this cut-off and the crank position when main valve closes, and prolong this bisector till it cuts the given circular locus, this bisector will constitute the limit to OD_{v_i} . A perpendicular dropped from the intersection D_{v_i} of this bisector with the circular locus will give the maximum lap ϵ_o that can be used in this gear with a negative direct valve. Such a lap will give the largest port openings, but if these are already ample a smaller lap ϵ_o may be chosen which will favor the average cut-off in the matter of rapidity of closing. In Fig. 100 and 101 a smaller lap than the maximum given above is used.

Let us again assume a new set of data in which the lap ϵ_o , the crank positions at the extreme cut-offs of the expansion valve and at the closing of the main valve are given, while the radius r_o of the expansion eccentric is required.

We know from the nature of the gear that r_o is constant and that O' is the center of the, as yet unknown, circular locus of D_v . Where the lap circle cuts the prolongations of the crank positions corresponding to minimum cut-off and closing of main valve, erect perpendiculars to the crank and find their intersection D_{v_i} . Then $O'D_{v_i}$ will be the radius of expansion eccentric which can be employed with the given lap ϵ_o .

The small laps of the expansion valve and the small throws of its eccentric in this case are due to the gridiron type of valve used. It is one of the good points of this engine that it thus

cuts down the work of valve friction without decreasing the excellence of the admission.

The cut-offs can be found more exactly from the accompanying diagram of piston travel, for which $\frac{L}{R}$ is again 6. (See method used in Fig. 63.)

MEYER VALVE GEAR.

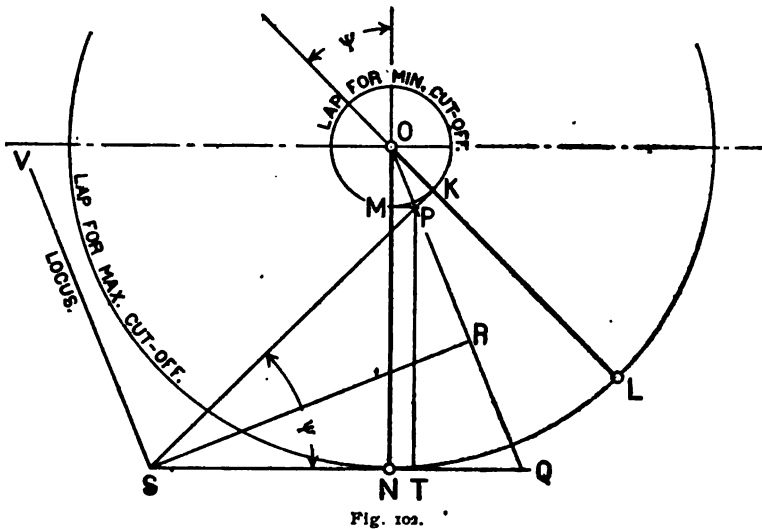
In this arrangement the expansion eccentric preserves an invariable throw and an invariable position relatively to the crank, the expansion being varied by changing the amount of the lap of the valve. The mechanism for doing this is shown in all the elementary text-books of the steam engine and concerning it we will simply say, that the expansion valve consists of two halves that may be moved apart to any desired extent by means of right and left hand screws located on the rod connecting these two halves. The separation may be carried so far as to change the valve from one having negative lap to one having positive lap. As the expansion eccentric does not change at all with the grade of expansion, there is in this case no locus of the center E_e of the expansion eccentric and then, strictly speaking, (in an existing gear), there is no locus of the center D_v of the virtual eccentric. The locus of D_v inscribed on Fig. 106 is simply to be regarded as a geometrical help in finding the angle of advance δ_v when the eccentricity r_v is known. As the expansion, like the main, eccentric is fixed as regards its setting and throw, there will be only one parallelogram of eccentricities (or of diameters), only one virtual eccentric and only one virtual valve-circle.

Before solving any problem in connection with this gear it will simplify matters to first demonstrate a couple of auxiliary propositions. The first of these relates to the finding of the locus D_v for a special case and the second to the equalization of the cut-off in the Meyer gear.

It can be shown that, if the difference $e_2 - e_1 = E$ is prescribed or given by the proportions of the expansion valve, the vertex D_v of diameter OD_v of virtual valve-circle must lie somewhere on a straight line SV , Fig. 102 and Fig. 106, parallel to the bisector OQ of the angle ψ ($= NOL$) included between the crank positions for maximum and minimum cut-offs, the distance of the locus from the bisector being

$$\overline{SR} = \frac{E}{2 \sin \frac{\phi}{2}}. \quad (91)$$

Let OL and ON represent the prolongations of the crank positions at the extreme cut-offs, OQ the bisector of the angle



$\psi = LON$ and $KL = MN = PT = E = e_2 - e_1 =$ given difference of outside laps. Then

$$\overline{SP} = \frac{E}{\sin \phi} \text{ and } \overline{SR} = \overline{SP} \cos \frac{\phi}{2} = \frac{E}{2 \sin \frac{\phi}{2}}.$$

The remarks and constructions given on pp. 194, 195 in connection with the Cummer engine, are also applicable here.

As equality of cut-off is conducive to smooth running, Dr. Burmester has given a solution of the problem of equalization for the Meyer valve, which consists in making the laps unequal for the two ends of the valve, the difference between the laps being a constant quantity.

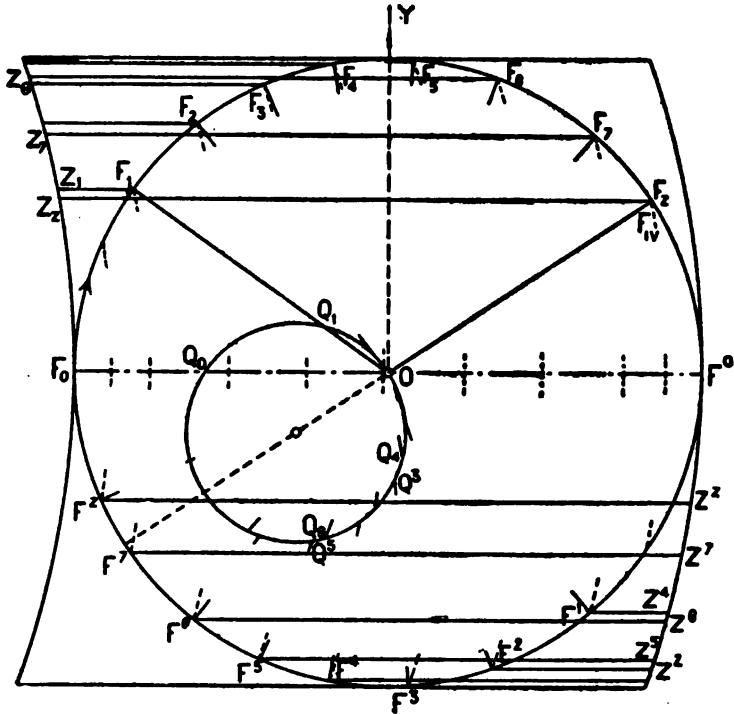


Fig. 103.

The procedure is as follows: In Fig. 103 the stroke F_0F_0' is divided into eight equal parts and through these divisions, with the connecting rod as radius, arcs are struck intersecting the crank circle at F_1, F_2, F_3 , etc., then $Z_1 F_1, Z_2 F_2, Z_3 F_3$, etc., represent $\frac{1}{8}, \frac{1}{4}, \frac{3}{8}$, etc., of stroke, or cut-offs, and OF_1, OF_2, OF_3 , etc., the corresponding crank positions. These cut-offs $\frac{1}{8}, \frac{1}{4}$, etc.,

are laid off as abscissas in Fig. 104. Now let the valve-circle for this case be represented by $OQ_0 Q_3$; it will cut from the crank the chords OQ_0, OQ_1 , etc.; these represent the travel of any point of the valve and are laid off as ordinates $T_0 m_0, T_1 m_1$, in Fig. 104.

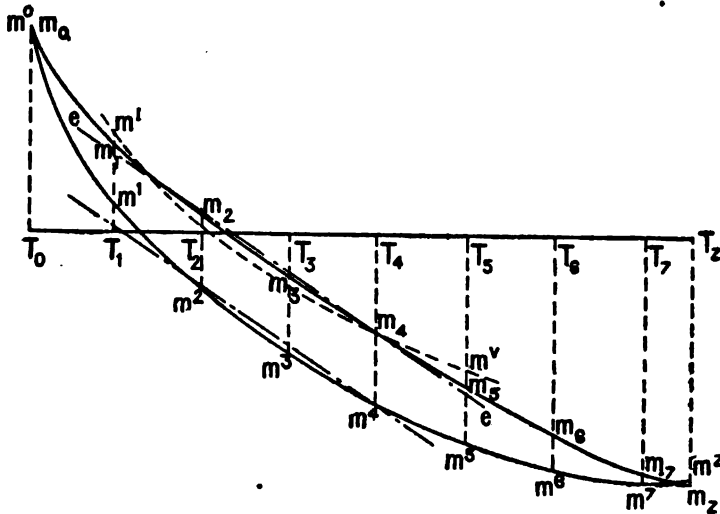


Fig. 104.

Each of these chords also represents the lap of the valve effecting the cut-off at the crank position on which the chord lies. Hence the ordinates of curve $m_0 m_1 m_2 \dots m_8$ also represent the laps effecting the cut-offs represented by their corresponding abscissas, the ordinates above the base representing positive, and those below negative, laps. In like manner the ordinates of curve $m^0 m^1 m^2 \dots m^8$ represent the laps effecting the cut-offs during the return stroke. It is evident from the figure that for the same cut-off the laps for the return stroke are greater than those for the forward stroke. In this diagram these differences can be made to (nearly) disappear by moving the lower curve upward through a distance $m^4 m_4$; it will then occupy the position shown by the dotted curve $m^1 m^7$. In the

mechanism itself, the corresponding step is to start with a symmetrical valve having the same lap on its two parts, then set the right-hand half of the expansion valve; nearer to its left-hand half by just this amount $m^+ m_+$. Approximately equal cut-offs will then be effected on each stroke.

It is evident from the diagram, Fig. 104, that a straight line ee will approximately represent the two lap curves between the cut-offs $\frac{1}{2}$ and $\frac{3}{4}$. To equalize the cut-off it is therefore unnecessary to move the two halves by two screws of unlike pitch. Both screws may be of the same pitch provided the two halves of the expansion valve are placed on their seat on the main valve with the unequal laps given above.

The method of finding the unknown elements in this (Meyer's) valve gear of course varies with the character of the data. We shall assume in all the special cases discussed, that the range of cut-off of the expansion valve is given and that the laps, setting and throw of the main valve are known. We therefore know the crank position at which main valve cuts off and care must be taken that the expansion valve does not permit a second admission of steam during the same stroke. Although the locus of E_e (and locus of D_e) is reduced to a point for this gear, it still remains true that the point D_e must lie at the distance r_o from the point O' ; this point, we know from Fig. 95, is found by prolonging OD_m and making $OO' = OD_m$. The problems and their solutions are arranged in the following order:

- (1.) Given r_o and δ_o ; required e_{oa} and e_{oi} .
- (2.) Given r_o and e_{oa} ; required δ_o and e_{oi} .
- (3.) Given r_o and $e_{oa} - e_{oi}$; required δ_o , e_{oa} and e_{oi} .
- (4.) Given δ_o ; required r_o , e_{oa} and e_{oi} .
- (5.) Given δ_o and $e_{oa} - e_{oi}$; required r_o , e_{oa} and e_{oi} .

(1.) Fig. 106. If the radius r_o and the angle of advance δ_o of the expansion eccentric are both given, the problem is readily solved, for then a parallelogram constructed on the main and expansion eccentrics (Fig. 92) will determine the virtual eccentric

and this (Fig. 60) the virtual valve-circle. The intersections of this circle with the crank positions corresponding to the limits of cut-off will determine the laps desired, provided the expansion valve does not open prematurely. In this gear, it is only at maximum cut-off that there is any danger of an opening of this valve before the main valve has closed the port at the cylinder end in question.

(2.) Fig. 106. Given r_o and e_{oa} . With O' as a center and r_o as a radius, describe a circumference; the vertex D_v will lie somewhere on it. At the cut-off point $1'$ erect a perpendicular $1'D_v$ to $O1'$, where it cuts the circular locus will be point D_v desired, provided the second intersection $2'$, of valve-circle on OD_v with lap circle e_{oa} , is such that the expansion valve's opening does not occur till after the closure of the port by the main valve, otherwise the data must be regarded as incompatible with good running. The lap e_{oi} can be found from the intersection of this virtual valve-circle with the crank position at minimum cut-off.

By means of the parallelogram, the position of OD_e may be found from OD_m and OD_v and then by method of Fig. 60 the position of OE_e . This determines at once the setting of the eccentric and by applying the rule given on p. 145 the angle of advance δ_o can be found if desired.

(3) Fig. 106. Given r_o and the difference $e_{oa} - e_{oi} = E$. We know from the remarks on p. 200 that D_v must lie somewhere on the circumference described with O' as a center and r_o as a radius, and from the proposition given with Fig. 102 we know that D_v must also lie on a line SV , parallel to bisector OQ (Fig. 102) of angle $\phi = \text{angle } 303' = \text{angle } 101'$ and at a distance from it equal to

$$\overline{SR} = \frac{E}{2 \sin \frac{\phi}{2}}.$$

In general there will be two intersections of this circumference and line SV , which will give two solutions of the problem. That one should be chosen which prevents a premature opening of the

expansion valve. In the present case $E = 1\frac{3}{8}''$, $r_o = 1\frac{3}{8}''$ and $\phi = 54^\circ$, then $\overline{SR} = 1.514''$. This gives two values of OD_v , $1.82''$ and $2.29''$; of these the former guards thoroughly against a premature opening for the whole range of expansion; the second value of OD_v permits premature opening for nearly the whole range of expansion. Of these two valves, $OD_v = 1.82''$ is decidedly the preferable one and is the one chosen for illustration in Fig. 106. By the aid of the parallelogram and the method of Fig. 60 we can now get the position of the expansion eccentric OE_e .

(4.) Given δ_o . This determines the location of the radial line OE_e of the expansion eccentric relatively to the crank OC , Fig. 106. Consequently (according to Fig. 95) the vertex D_v of the virtual eccentric must lie on the line $O'F$, the position of this line relatively to a vertical and horizontal axis through O' being the same as the position of OE_e relatively to a vertical and horizontal axis through O .

The bisector of the angle $1'02'$, between crank position when main valve cuts off and the crank when expansion valve effects its maximum cut-off, will intersect line OF' in a point which is a limit of the possible vertices D_v on $O'F$. The intersections, of the valve-circle described on OD_v as a diameter with the cut-off positions, will give the maximum and minimum laps desired. The distance $O'D_v$ will be equal to the eccentricity r_o of the expansion eccentric.

(5.) Given δ_o and the difference $e_{o_1} - e_{o_2} = E$. As in case (4), we find $O'F$ to be one locus of D_v . The construction of Fig. 102 gives a second locus SV of D_v . The intersection of these two loci will give $O'D_v = r_o$, provided D_v is located on, or to the left of, the bisector of the angle $1'02'$. Perpendiculars dropped from D_v on to the crank at the extremes of cut-off will then give e_{o_1} and e_{o_2} .

Fig. 107. The Rider Valve gear is only a particular case of the Meyer. In the Rider the expansion valve plate instead of

being flat, as in the Meyer, is bent into a cylindrical shape. This cylinder is rotated by a governor when the grade of expansion is to be changed, the rotation effecting the necessary change of lap. The principle of this construction is shown in Fig. 107, where the cylinder's surface and its seat are both developed into a plane. Here the corresponding edges of the two ports are inclined to each other but are parallel to the edges of the valve. When the latter is in the position shown by the full lines the cut-off = 0.5, the lap is negative and equal to 1.67"; when the value has been shifted through fe into the position indicated by the dotted lines the cut-off is 0.1 and the negative lap = 0.30, the difference of the two laps being $fg = 1\frac{3}{8}"$. The amount of shifting ef in the flat plate is equal in the cylindrical valve (with helical edges) to the circular arc through which any point on its surface is moved. In order that the valve may take up some of the wear between its face and seat, this circular seat should be considerably less than 180° ; a value of about 100° will be found to be suitable. The proportions and distribution given in Figs. 105-107 are suitable for a Rider valve gear. A Meyer gear can easily be arranged so as to cut off from 0 to $\frac{1}{4}$ ths.

RIGG ENGINE, FIGS. 108, 109, 110.

(See Rigg's Treatise on the Steam Engine, Pls. 17, 18, pp. 95-97).

In this engine the expansion is varied by a changeable eccentricity, the setting of the eccentric remaining the same at all grades of expansion. The centers E , of the expansion eccentric must therefore lie on the same radial line OE_1E_2 , Fig. 109, and the center D_v of the virtual eccentric must likewise lie on one straight line $O'D_vD_2$, whose location with respect to the axes $X'O'Y'$ is like the location of OE_1E_2 or OD_1D_2 to the axes XOY (see also Fig. 95.)

We again assume the extremes of cut-off as given and the limits within which the expansion valve may open without per-

mitting a second admission during the same stroke. The laps, setting and throw of the main valve are also known. As these data are like those for the Cummer gear the remarks on p. 194 concerning the limitations to which the location of OD_v is subjected apply here also.

We shall discuss the problems connected with this gear as follows:

- (1.) Given e_o , required δ_o , r_{o_1} and r_{o_2} .
- (2.) Given δ_o , required e_o , r_{o_1} and r_{o_2} .
- (3.) Given $r_{o_2} - r_{o_1}$, required δ_o , e_o , r_{o_1} and r_{o_2} .

(1.) Given e_o . This case is solved in Fig. 109. JO and KO are the crank positions corresponding to minimum and maximum cut-off, respectively. $OH = OI = e_o$ is the given, negative, lap of the expansion valve. Here the diameter OD_{v_1} of the virtual valve-circle for minimum cut-off is taken as located at one of its limits, namely that one which corresponds to a reopening of the expansion valve at the very instant that the main valve closes. It would have been better to have assumed some margin. Position OD_{v_1} is therefore a bisector of the angle included between the prolongation of crank position JO and the prolongation of crank position corresponding to the closing of the cylinder port by main valve. A second locus of D_{v_1} is the perpendicular at H to JOH ; the vertex D_{v_1} lies at the intersection of this bisector and this perpendicular. For this kind of gear all the points D_v lie on a straight line through O' , therefore by joining D_{v_1} and O' we get a locus of the unknown vertex D_{v_2} . A second locus of D_{v_2} is obtained in the perpendicular erected at I to the crank position KOI for maximum cut-off. We have $O'D_{v_1} = r_{o_1}$ and $O'D_{v_2} = r_{o_2}$; the remaining unknown element δ_o can be found by means of the parallelogram and Fig. 60, as in the preceding valve gears, or more directly as indicated in the figure. The crank is supposed to be at OX when the expansion eccentric is on radius OE_1E_2 .

- (2.) Given δ_o . This is the preceding case worked backward.

We therefore use the same figure, 109. As the locus OE_1E_2 of the expansion eccentric is given we find at once OD_1D_2 and its parallel $O'D_{v_1}D_{v_2}$. The latter is the locus we need and its intersection D_{v_1} with the limiting position of diameter OD_{v_1} must be projected on crank position JOH to get the lap $OH = e_0$. With this lap we can now easily find OD_{v_2} . Then will $O'D_{v_1} = r_{o_1}$ and $O'D_{v_2} = r_{o_2}$.

(3.) Fig. 110. Given $r_{o_2} - r_{o_1}$. In this case we do not know either of the eccentricities of the expansion eccentric at the cut-off limits, we simply know the *change* of eccentricity which our mechanism will permit. We again assume, for the location of the diameter of virtual valve-circle corresponding to minimum cut-off, the limiting line Ol , Fig. 110.

We can easily find a locus for the vertex D_{v_2} belonging to maximum cut-off. For this purpose draw through O' a series of lines $O'L, O'M, O'N \dots$, which cut line Ol in points $l, m, n, \dots$; then lay off distances $lL, mM, nN, \dots$ each equal to the given variation of eccentricity $r_{o_2} - r_{o_1}$. The curve L, M, N, D_{v_2} thus obtained is part of a conchoid and is the locus mentioned. Each point on this curve represents a different maximum cut-off, for the constant difference $r_{o_2} - r_{o_1}$, and our task now is to find the point corresponding to the given maximum cut-off. This is most easily accomplished by a trial process, which we will now give. Inspection of Fig. 109 for the preceding cases, shows that the intersection C of the two perpendicular CH and CI dropped from the vertices D_{v_1} and D_{v_2} , falls on the line GC bisecting the angle JOK included by the crank positions at the cut-off limits. This suggests the following tentative method: Reproduce, on a separate piece of tracing cloth, (see right half of Fig. 110), the bisector $G'C$ and the perpendiculars $C'I'D''$ and $H'C'D'$ so that angles $G'C'D'' = GCD_{v_2}$ and $G'C'D' = GCD_{v_1}$. Then place this tracing on the drawing so that $G'C'$ on tracing will coincide with GC on drawing and slide bisector $G'C'$ on bisector GC till the intersection D_{v_1} of perpen-

dicular $C'D'$ with line OD_{v_1} and the intersection D_{v_2} of perpendicular CD'' with conchoid LMN both fall on the same straight line $O'D_{v_1}D_{v_2}$ drawn through O' . When this occurs all the conditions of this particular case are fulfilled and the size and setting of expansion eccentrics can be found in the now well-known manner.

Before leaving this problem it should be noticed that the locus of the vertex D_{v_2} , (of the diameter OD_{v_2} of the valve-circle corresponding to a given maximum cut-off), is an hyperbola $ROD_{v_2}Q$, whose asymptotes PS and PT are parallel respectively to the perpendicular CID_{v_2} and the diameter OD_{v_1} . The origin O is on the one branch of this curve and O' on the other branch. Points on the hyperbola are easily obtained as follows: Take any point C on the bisector GC and drop the perpendicular CH and CI on JH and KI . Then prolong HC till it cuts given line $O'I$ in some point, say D_{v_1} . Join this point with O' and prolong the connecting line till it cuts the second perpendicular CI in D_{v_2} . This will be a point on the hyperbola. It is evident that the intersection of conchoid and hyperbola give at once exactly the desired point D_{v_2} . But the construction of both these curves, or even partial arcs of both of them, will seldom be advisable, as the problem can be much more easily solved by the trial method detailed above.

In the preceding double-valve gears only one of the elements, (lap, setting and throw) has been varied at a time. In the remaining gears, two elements, or their equivalents, are varied simultaneously.

STURTEVANT ENGINE, 12x24, FIGS. 111, 112, 113.

In this engine the expansion is varied by a simultaneous variation of the throw and angle of advance. This variation is effected by a suitable mechanism, of the swinging eccentric type, the center of the expansion eccentric being compelled to travel on the circular locus E_e (Fig. 112) when the expansion is to be

varied. This locus passes O at a distance of $\frac{3}{4}''$. Its center and radius are inscribed on the figure. According to figure 95 the locus of D_v has, relatively to O' , the same position as locus E_v to O .

Here also we assume laps, setting, throw and distribution of main valve as completely determined. The range of cut-off for expansion valve, and the limits of the opening of this valve are likewise supposed to be known. On the diagram of Fig. 112 the minimum cut-off is stated to be 0. This is approximately true but not exactly. A tangent touching both lap circle AO and locus of D_v will determine, at its point of contact A , the crank position AOC for least cut-off.

To find the position of expansion eccentric at 0 cut-off, we erect at A (Fig. 112) the perpendicular AD_{v_1} ; it will touch the locus of D_v in D_{v_1} and OD_{v_1} will be the diameter of the virtual valve-circle. The dotted parallelogram $OD_{v_1}D_{e_1}D_m$ gives the relative position and magnitude of the diameters of main, expansion and virtual valve-circle; the equal parallelogram $OE_{v_1}E_{e_1}E_m$ gives the relative position and magnitude of the corresponding eccentrics. $OE_{e_1} = r_o$, is the position of the expansion eccentric relatively to crank OC when cut-off = 0.

Fig. 113. In like manner we find that OD_{v_2} is the diameter of the virtual valve-circle for the maximum cut-off = 0.7. The reopening of the port by expansion valve occurs just after main valve closes. For this cut-off the position and size of expansion eccentric is given by OE_{e_2} , when crank is at dead point OC . The expansion eccentric therefore travels on locus from OE_{e_1} to OE_{e_2} , while varying the expansion.

DOUBLE-VALVE GEARS WITH INVARIABLE DRIVING ECCENTRICS.

There are two kinds of double-valve gears in which expansion valves are driven by non-adjustable or invariable eccentrics. In the first kind the expansion valve is driven by one or more invariable eccentrics, and its absolute motion is entirely indepen-

dent of the main eccentric. In the second kind the absolute motion of the expansion valve does depend, more or less, upon the main valve's eccentric. In both kinds there is between the expansion valve and the invariable eccentric a mechanism whose action is equivalent to driving this valve by a variable eccentric.

A valve gear by G. Schuhmann, of Reading, Pa., belongs to the first kind. Between the expansion eccentric and its valve there is a rocker with two arms, a variable and invariable one, the invariable one driving the expansion valve and the variable one receiving its motion from the expansion eccentric through a block whose position on the arm is regulated by the governor. The locus of the center E_e of the expansion eccentric is a straight line which, when crank is on back center, passes below and to the right of the shaft center O making an angle of 30° with the line of dead centers. The main valve in this engine belongs to the positive indirect, and the expansion valve to the negative indirect, type.

The valve gears of Polonceau (or Borsig), of Guinotte and of Bilgram are examples of the second kind. Polonceau's is described and discussed on p. 203, second English edition, of Zeuner's Valve gears. Here both main and expansion valves are driven by Gooch's link motion and the locus of the center E_v of the virtual eccentric is a vertical line through center of shaft. Of these gears we will discuss only those of Guinotte and Bilgram.

GUINOTTE'S VALVE GEAR, FIGS. 114, 115, 116.

The complex mechanism and theory of this gear are both given by Dr. Zeuner (see p. 214, second English edition of Valve Gears). It permits the reversal of engine without touching the expansion gear. The character of the locus of the vertex of the diameter of the virtual eccentric is such as seems to deserve special treatment here. We therefore reproduce it although it is

fully given by Dr. Zeuner. In this gear the vertex D_v of the diameter of the virtual valve-circle lies in a rectilinear and inclined locus NN , Fig. 115. This is similar to the locus MM of the corresponding vertex D_e of the expansion valve-circle and this in turn corresponds to locus LL of the center of the *equivalent* expansion eccentric. We say *equivalent* because the actual expansion eccentric has a constant eccentricity $OD_o = r_o$ and a constant angle of advance, $\delta_o = 90^\circ$. But the mechanism between it and the expansion valve has the effect of moving the latter as if it were driven by an expansion eccentric whose center had a locus LL for the different grades of expansion. According to the principle underlying Fig. 95, the position of NN relatively to O' as an origin is the same as that of MM relatively to O as an origin and the same as LL relatively to O .

At position 3, (Fig. 115), the main valve cuts-off at the left cylinder port and in order that the expansion valve shall shut its left (main valve) port at the same time we must give the diameter of the virtual valve-circle a position OD_{v_2} , which is found by erecting, at the lap-circle I , the perpendicular ID_{v_2} to the cut-off position $3O1$. The distance OD_{v_2} will be the corresponding diameter of expansion circle and OE_{v_2} the corresponding position of the *equivalent* expansion eccentric. In the mechanism itself this corresponds to the position of the block at the upper end of the link. It is evident that with negative lap the virtual valve-circle OID_{v_2} will cut-off at the same instant that the main valve closes.

Fig. 116. For a minimum cut-off we will choose 0 or cut-off at dead point. To find the corresponding virtual valve-circle we erect to the dead center line, at the lap circle H , the perpendicular HD_{v_1} and prolong it to the locus NN of the vertex D_{v_1} . The distance OD_{v_1} will be the diameter of the virtual valve-circle OHD_{v_1} . It is evident that this gives the desired cut-off and that the reopening of the left port in the main valve will not take place till crank reaches position $O4$, that is after the main valve

has closed the left cylinder port. The position of the equivalent expansion eccentric will be near OD_e , in this case and in the mechanism it will be found that this requires that the block shall be at the bottom and not at the central portion of the curved slot in the link.

BILGRAM VALVE GEAR, Figs. 117, 118, 119.

The mechanism of this gear has been given on p. 119 of Bilgram's work on Slide Valve Gears, also in Halsey's Valve Gears, p. 123. There is only one eccentric which drives the main valve directly and the expansion valve indirectly.

The inventor gives the favorable proportions, for most of which the reader must look to the references just given. The mechanism between the main eccentric and the expansion valve, in its action on the latter, is nearly equivalent to the action of an expansion eccentric whose center travels on the circular locus OLE_e , Fig. 118, while passing from one extreme of expansion to the other. A line $\overline{OE_e} = 2 \times OE_m = 2r$ placed 50° in advance of the main eccentric radius $\overline{OE_m}$ is the diameter of this locus. The corresponding circular locus OMD_v of the vertex D_v of the diameter of the virtual valve-circle is found in accordance with the principle illustrated in Fig. 95.

Fig. 118. The inventor recommends a negative lap for expansion valve that is $\overline{\quad}$ than the outside lap of the main valve. We have here taken $\frac{1}{2}''$ for this negative lap because this is favorable to quick cut-off. To get a cut-off at the dead point erect to OH the perpendicular HD_{v_1} and prolong it till it cuts the locus $OMD_{v_2}D_{v_1}$ in D_{v_1} . Then will $OH D_{v_1}$ be the required virtual valve-circle for minimum cut-off. It is evident that the reopening of the left, main-valve, port, occurs at position $4OE$ after the main-valve has closed the corresponding cylinder port.

Fig. 118, 119. The cut-offs at one quarter and one half stroke are treated in the same way; the corresponding virtual

valve-circles are OID_{v_2} and OJD_{v_3} (Fig. 119). Inspection of these circles at the points I and J shows that the cut-off is very quick, as is claimed.

The range of cut-off is great, extending from nothing up to that effected by the main valve. The virtual valve-circle corresponding to this maximum cut-off is OKD_4 , Fig. 119.

PRACTICAL DATA AND DIRECTIONS.

The rapidity of cut-off for any gear depends upon the speed of the valve at the instant and the number of ports through which the steam is admitted to one end of cylinder. It can easily be estimated by finding the position of the virtual eccentric at this instant and dropping a perpendicular from its center upon the "dead point" line of the eccentric. The product of the number of ports and the ratio of this perpendicular to the perpendicular dropped from crank-pin on piston stroke is, other things being equal, perhaps as fair and convenient as any for comparing the rapidity of cut-off of the different gears.*

The distance from the port of a valve edge not engaged in steam distribution should be more than sufficient to prevent its opening the port at maximum travel.

To avoid two admissions of live steam during the same stroke the expansion should not reopen its port in the main valve till after the main valve has closed the corresponding cylinder port.

Arrange the valve gear so that it will realize the steam distribution required by the desired indicator diagram. The release

* A more exact measure would be the product of the number of ports by the ratio of valve speed to piston speed at instant of cut-off. The valve speed is measured by the intercept included between the shaft's center and the eccentric rod, the intercept being taken on the perpendicular through this center to the stroke of the valve end of the rod. The piston speed, to the same scale, is measured by the intercept on the perpendicular to its stroke included between shaft center and connecting rod. The ratio of valve speed to piston speed is the ratio of these two intercepts.

may begin at $\frac{1}{8}$ th of the stroke, unless it has already been fixed by the compression assumed in the earlier part of the work. As American practice now is to use balanced valves, large throws of eccentrics are permissible. The cut-off should be made as rapid as possible.

The width of bridge (metal between steam and exhaust ports) should be greater than $r - (b + e)$ to avoid fresh steam passing directly into exhaust. It is often made equal to the thickness of the cylinder wall. An empirical formula given by Zeuner is $0.4 + 0.5b$.

To prevent steam leaking between valve face and seat they should have a contact-width of at least 0.4 of an inch.

The exhaust port should have such a width b_o , that when valve is at end of travel, it will be still have the width b , of the steam port, that is, $b_o = r + b + i - (0.4 + 0.5b)$
 $= r + 0.5b + i - 0.4$.

The valve seat should have such a length that the valve will pass beyond the outer edges of the seat, thus avoiding the formation of a shoulder in the seat. There should always remain however a 0.4" contact to prevent leakage.

When cut-off ϵ , lead v and maximum port-opening $b \pm k$ ($+k$ = over-travel, *i. e.*, beyond edge of port) are given, the half throw r can be found by trial, or at once and exactly, by Zeuner' formula:

$$r = \frac{2(b \pm k) - v + 2\sqrt{(b \pm k)(b \pm k - v)(1 - \epsilon)}}{2\epsilon} \quad (92)$$

The following tables will be useful in the solution of problems relating to the main valve. When the outside lead is known as a fraction of the eccentricity r , for example $v = mr$, we may transform the formula into

$$\frac{b + k}{r} = 1 + \frac{m}{2} - \sqrt{1 + \frac{m^2}{4} - \frac{m^2}{4\epsilon} - \epsilon}. \quad (93)$$

VALUES OF $\frac{r}{b+k}$.

Values of $\frac{v}{r} = m$	Values of $\epsilon =$ apparent cut-off.								
	.50	.55	.60	.65	.70	.75	.80	.85	.90
0.05	3.140	2.821	2.545	2.307	2.094	1.905	1.730	1.567	1.410
0.075	3.018	2.721	2.464	2.240	2.039	1.859	1.693	1.537	1.386
0.10	2.906	2.626	2.387	2.176	1.987	1.815	1.657	1.508	1.362
0.125	2.792	2.536	2.314	2.116	1.937	1.774	1.622	1.479	1.339

VALUES OF $\frac{\epsilon}{r}$.

Values of $m = \frac{v}{r}$	Value of $\epsilon =$ apparent cut-off.								
	0.50	0.55	0.60	0.65	0.70	0.75	0.80	0.85	0.90
0.05	.681	.645	.607	.566	.522	.475	.422	.362	.291
0.075	.669	.632	.594	.554	.510	.462	.409	.349	.278
0.10	.656	.619	.581	.540	.497	.449	.396	.337	.266
0.125	.642	.606	.568	.527	.484	.436	.383	.324	.253

For any assumed value of $\frac{r}{b+k}$ we can immediately get the angular advance from

$$\sin \delta = \frac{\epsilon + v}{r} = 1 - \frac{b+k}{r} + \frac{v}{r} = 1 + m - \frac{b+k}{r}. \quad (94)$$

k should be taken as small as is consistent with sufficient port opening. The shorter cut-offs given in table are more favorable than the longer ones in preventing the reopening of the expansion valve before main-valve closes, on the other hand the shorter cut-offs involve a greater throw of main valve.

In slow going engines the outside lead is often taken $= \frac{1}{18} r$, while in fast running engines $\frac{1}{8} r$ is often taken.

VALVE GEARS WHICH DO NOT POSSESS "HARMONIC" MOTION.

On p. 134 we divided all valve gears into two classes: those in which the valves moved approximately like the "slotted cross head" and those which did not, that is, into valves with and without "harmonic" motion. It is the first class that is most extensively used in practice and the one which has been discussed in the preceding pages. Even in these there is some deviation from "harmonic" motion which is not due to the angularity or length of the eccentric rod. As the valve stem is often at a considerable distance from the plane of the eccentric, sometimes at the side, and at other times on top, of the cylinder, rocker arms are interposed between eccentric and valve to transmit the motion. Usually they produce less deviation from "harmonic" motion than the angularity of the eccentric rod. But they may be arranged so as to produce a rapid opening of the port followed by a period of comparative rest for the valve. The Corliss valves driven by a wrist-plate and the Porter-Allen valves driven by the bent lever $O'R$, PQ' in Figs. 80-82, are examples. Mr. F. A. Halsey has shown, in his Slide Valve Gears, pp. 54-63 and 89-98, that the rocker arms may be arranged so that both cut-off and lead are equalized. In the Ide engine they are used to keep the lead constant at each end but greatest at the cylinder end farthest from shaft. All such devices produce some deviation from "harmonic" motion.

The most characteristic members of this group are the so-called "Radial" valve gears, of which the Joy is an excellent representative and perhaps the one which is best known in this country. The link driving the valve rod in these gears has two of its points guided in curves, one of which is closed and the other open or closed, while a third point drives the valve rod or stem. The motion is usually of a complex character and the valve travel is got at by plotting the paths of the three points of the driving link, in much the same fashion as the travel of the Porter-Allen valves was obtained. The relation between valve

and piston travel may also be shown by polar or oval diagrams like those given Figs. 83-91 for the Porter-Allen valve gear.

This group of non "harmonic" valve motions, (see p. 134), has these two following subdivisions:

(a.) Gears in which the valve slides on a stationary seat and only its absolute motion is considered.

(b.) Gears with two valves, of which one slides on the other, and in which the relative as well as the absolute motions of the valves must be considered.

The Porter-Allen motion, already discussed, belongs to the first subdivision and may be regarded as composed of three "single-valve" gears, each valve sliding on a fixed seat.

The Payne-Corliss engine belongs to the second subdivision. It has two sets of double-valves, one set for each end of the cylinder. Between the eccentric and the valves there are Corliss wrist-plates, but not the Corliss releasing-gear.

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